HEATING SYSTEMS

AND THE DESIGN OF
HOT WATER AND STEAM HEATING
APPARATUS

F. W. RAYNES

DEPT. OF ARCHITECTURE
DEC 4 . . 1913
UNIVERSITY OF TORONTO

HEATING SYSTEMS

BY THE SAME AUTHOR

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HEATING SYSTEMS

DESIGN OF HOT WATER AND STEAM HEATING APPARATUS

BY

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PREFACE

This work presents the most modern practice in an important branch of Engineering. It is freely illustrated with a large number of drawings, and some catalogue prints are also included where special drawings would possess little further advantage. It is not, however, the writer's intention to convey the impression that he favours the products of one maker in preference to those of another.

A special feature of the book is the large number of Charts that have been prepared, and the method adopted in sizing the pipes of different systems. Attention has been directed rather to the practical than to the theoretical aspects of the work, whilst in the sizing of pipes, the process in a large measure is merely a mechanical one. Consideration is also given to the economical aspect of heating problems, and especially in connection with the heating of works or of industrial buildings. The book is therefore intended for the busy professional or business man, as well as for the use of students.

The writer desires to thank all who have assisted either directly or indirectly in the production of the work.

F. W. R.

The Royal Technical College,
Glasgow.

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CHAPTER I

GENERAL

On reviewing the development of heating and ventilating apparatus, one cannot fail to be struck with the progress made during the last few years. On the economical side of heating work, much, however, remains to be done, in order that a large proportion of the heat now lost can be utilized for the service of man.

Ventilation.—The advantages derivable from a plentiful supply of pure fresh air at a suitable temperature and humidity are widely recognized, although there is a sharp difference of opinion as to the precise form a plant should take to give the best results.

In Great Britain there has been no legislation up to the present to enforce the ventilation of public buildings. This is most unfortunate; for the heating and ventilating of these places are often notoriously bad. In a comparatively new school the writer inspected a short time ago, less than 5500 cubic feet of air were flowing through the fresh-air inlets of one of the rooms per hour. This was often occupied by fifty-four children. Thus less than 100 cubic feet of air per hour per person were accounted for by direct ventilation. Although this would scarcely represent the true state of affairs owing to air leakages, the case is sufficiently marked to show that little wonder need be expressed at the difficulty experienced by the children in performing their tasks, on account of the drowsiness produced by the stuffy and very humid atmosphere that envelops them. This case, of course, is aggravated in the summer time.

The ventilation of many other buildings is as bad as the case cited, and it is high time a certain minimum standard was enforced by law in the interest of the community at large.

In most of the American States 1800 cubic feet of air per person per hour is the minimum volume allowed for public buildings, and this is generally regarded as the smallest volume that should be provided to keep the atmosphere of a room reasonably pure.

Impurity of Air.—On some phases of ventilation opinions are undergoing a change. For example, the generally accepted standard of air, as regards its fitness to be inhaled, has been based upon its chemical impurity in terms of the contained carbonic acid gas. Roughly speaking, it has been considered important that in ten thousand parts of air there should be not more than twelve parts of carbonic acid gas, whilst for good ventilation the same volume of air should not contain more than six parts of carbonic acid gas.

Many physiologists now maintain that the percentage of carbonic acid gas is of no great moment, and the thing that really matters is the air temperature and percentage of moisture present. Quite recently, experiments on the physiological aspect of ventilation have been conducted, both here and abroad, the general conclusions being the same. The method adopted in these experiments has been to confine one or more subjects in a small chamber, the ventilation of which is under absolute control. During the period of confinement the carbonic acid gas has been allowed to run up to over one hundred parts in ten thousand parts of air, and in some cases to over twenty times the amount that is likely to accumulate in ordinary rooms. It is stated that under these conditions the subjects experienced no discomfort.

Of the experiments made by Dr. Leonard Hill, of the London Hospital, the following observations were made by him some time ago to the Royal Commission appointed to inquire into the condition of weaving sheds :—" We have a small chamber which holds about 3 cubic metres. Into this chamber I put eight of my students, and seal them up. . . . At the end of half an hour the wet-bulb temperature has gone up to 85 degrees " . . . when " their faces are congested with blood. The CO_2 has gone up to four, or even five, per cent., and the oxygen down to a corresponding extent. Well, in these conditions I put on three electric fans, and do nothing else than

whirl the hot air just as it is . . . the students at once feel as comfortable as possible, but immediately the fans are stopped they feel as bad as ever, and beg for the fans to be started again. . . . All this tends to show that when you have a stationary moist air, warmed up around the body, you get discomfort; when the fans are put on the air is stirred up, and the cool air is brought into contact with the body, and the discomfort ceases."

In quoting a few passages apart from their context, the writer does not wish to convey the idea that the doctor is not an advocate of a plentiful supply of air, but that he, along with other physiologists, contends that the poisonous effects of carbonic acid gas have been overstated, whilst insufficient attention has been paid to the conditions of temperature and humidity.

For many years those interested in the problem of ventilation have not considered a specific quantity of carbonic acid gas in itself as injurious, but rather as an index of the contained organic impurity which may be harmful.

Organic Poisons.—It has generally been assumed that expired air contains matter of a poisonous nature, although on this point physiologists differ. Some experimenters, to prove there are toxic constituents in expired air, have collected liquid from the condensed vapours of expired air, and injected it into animals. The subjects of the experiments in some cases succumbed, but whether it was due to poisons or other effects must be left to the physiologists to decide. Other experimenters have failed to detect organic poisons in expired air, and in an address to the British Institute of Heating and Ventilating Engineers, Dr. Leonard Hill expressed the opinion that "the experiments which started this organic poison theory are really absurd. I cannot find that this organic poison exists."

Assuming the question of organic poisons in expired air is a disputable one, there is no doubt as to the presence of infective bacteria when people have colds, and other infectious complaints. It is, of course, impracticable to have an atmosphere free from infectious germs; but the more effective the ventilation, or the purer the air, the less risk there is of infection, and where other things are equal, the better able the constitution becomes to resist the attack of poisonous germs.

Important as ventilation is, immunity from disease does not by any means entirely depend upon it. Suitable diet, sunshine, rest, and environment all play their part, as well as psychological and physiological factors.

From the experiments of Dr. Hill and others, it has been shown that a person may live in stale air for a prolonged time, without experiencing discomfort or apparently any ill effect, so long as it is kept circulating, and the temperature and moisture do not rise too high. Very few, however, would be satisfied with such an atmosphere because injurious effects had not been proved. If a person is run down in health a visit to some health resort, or a sea voyage, is often suggested; the consumptive has outdoor treatment. In each case, the intention is for the subject to be more or less continuously swept by a pure cool atmosphere. This is the idea to be incorporated in a system of ventilation, as advanced by the modern physiologist.

The problem of ventilation is not always a simple one, for it is not merely a matter of passing so many cubic feet of air into a building.

When people congregate in large numbers in a room the temperature of the latter is raised to an appreciable extent through the heat evolved by them. As soon as the air temperature approaches that of the body, the radiation of heat from the occupants is interrupted, whilst the circulation of air about them is retarded. This has the effect of causing the occupants to be enveloped in a very humid and stagnant atmosphere, and this it is that causes the drowsy feeling and discomfort so frequently experienced in a large audience.

Ideal System.—The ideal ventilating system is sometimes said to be the one that will reproduce natural conditions. This is not a practicable thing, even assuming the difference in the chemical purity of internal and external air to be ignored. External conditions are very changeable, and to these humanity adapts itself. When, however, it comes to internal temperatures, our civilization requires these to be adjusted to suit the individual, although it is the antithesis to the natural order.

The maintenance of uniformity of temperature in buildings has received considerable attention, and many special appliances have been devised to attain this end. From a physiological

standpoint this side of the problem may have been carried too far, but the saving effected in fuel by the prevention of overheating has been greatly in favour of the installation of these fittings.

As regards the purity and suitability of the air that can be obtained in large structures this is very much a problem of cost for the handling and treatment of the air, the means adopted for its distribution, and the temperature at which it is admitted.

Ozone.—Another phase of ventilation is the use of ozone which is generated by a special machine. As an adjunct to a ventilating plant this has advantages for deodorizing purposes, and for imparting freshness to air when for economical reasons sufficient air changes cannot be effected. For ventilation, ozone is only used in a very diluted form, a high concentration being injurious in that it acts as an irritant on the respiratory tract.

As ozone is a very unstable quantity, its efficacy in ventilation will largely depend upon how it is introduced and upon the temperature of the air. If the air is ozonized and afterwards brought in contact with heating surfaces, it is possible that very little ozone will pass with the air into an apartment. More or less dust always gathers upon and floats about the heating surfaces, and with this the ozone combines, and all the more readily when it contains matter of an organic nature and is in a heated state. The use of ozone, in the writer's opinion, is better where the air does not require to be warmed.

On a large scale ozonized air is being used by the Central London Railway Company, and when the scheme is complete, it will be capable of delivering 80,000,000 cubic feet of ozonized air per day through the tubes and into the stations. The proportion of ozone used is one part in ten million parts of air.

Heating and ventilating may be carried out either as separate units, or they may be combined. Each method has its special features according to the class of structure to be dealt with.

Natural Ventilation.—Natural or gravity ventilation, although suitable for small and one-storey buildings that are not very wide, is inadequate for large high buildings. As the term implies, it is entirely dependent upon the elements or forces of nature, and

as these are erratic in their action, the systems depending upon them act in the same way.

A combined arrangement of mechanical and gravity ventilation, if properly designed, will satisfy most conditions that may be demanded. The air before delivery into buildings may be purified, tempered, or cooled, its relative humidity adjusted to suit the changing weather conditions, a definite volume of air may be delivered to any point desired, and a system as a whole may be easily controlled.

Mechanical Systems.—In mechanical ventilation the air moved is frequently utilized as the conveyer of heat for warming purposes, and such systems are often designated as "hot blast" ones.

In America, hot-blast heating for public buildings, as generally carried out, is looked upon by many with disfavour. The high temperatures to which the air is usually raised bring about chemical changes by decomposing some of the contained dust. Thus the air loses its freshness, and becomes to a certain extent vitiated before it enters a room. It is also thought that the overheating process devitalizes air by robbing it of properties that cool fresh air contains.

So far as Great Britain is concerned, the drawbacks of hot-air heating have never been so acute as in America. This, however, is not due to the superior way in which the engineering side of the problem is handled, but rather to our mild climate, and to the lower temperatures to which we are accustomed.

Good results can be obtained by "hot-blast" or indirect heating, but to get these the temperature of the heating surfaces and that of the air must be kept down, whilst the heating surfaces should also be kept as free as possible from dust.

In large buildings independent heating and ventilating is now very much practised, direct heating surfaces being used to replace the heat lost by windows, walls, and other cooling surfaces. The air, under these circumstances, only requires to be raised at the fans to a temperature a little higher than that maintained in the rooms. Such an arrangement permits of the fans being stopped when a building is unoccupied, power is saved, whilst the temperature may be fully maintained by the direct heating surfaces; further, it is very flexible in operation,

and the volume of air to any apartment may either be diminished or increased without affecting its temperature.

Hot-blast systems of heating and ventilating possess advantages for industrial buildings, in that the whole of the heating surfaces and power units are centralized. The initial cost is also lower than where the heating and ventilation are treated separately. There is no one system that will satisfy the requirements of all classes of buildings: every case requires to be considered on its own bearings.

Mechanical ventilating plants are usually classed either as "vacuum," "plenum," or combined ones. Each system has its advantages and limitations. A "vacuum" system is understood to be one where the air is withdrawn from an enclosed space by locating fans in the outlet ducts, and where the air pressure of the space is slightly less than that outside.

Vacuum Systems. Generally speaking, when a vacuum system is installed in an ordinary building, its success depends upon the location, number, and area of the fresh-air inlets. Badly designed vacuum systems are liable to produce unpleasant draughts, owing to the inward leakage of air around windows, doors, and through other crevices; the air currents may move directly from inlets to outlets without being diffused over the greater area of a room.

For localized ventilation, a vacuum system is specially valuable, such as for the immediate and direct removal of dust and fumes that are produced in connection with dangerous trades. Smoke rooms and other apartments into which vapours are emitted can be better ventilated by a vacuum than by a plenum system.

Plenum systems of ventilation are those in which the air is propelled into rooms by locating fans or other air movers in the inlet ducts. It is assumed in these systems, that the air in the apartments exceeds in pressure the external atmosphere, and that any air leakage through irregular channels will rather be outwards than inwards. Whether this will be the case or not largely depends upon the features embodied in the design. That considerable outward leakage takes place in many plenum systems is well known, owing to the fact of the entering air sometimes exceeding by 25 per cent. that recorded at the

outlet. On the other hand, a number of systems admits of more or less considerable inward leakage.

The principal merits attributed to plenum systems are, less liability to draughtiness, and the better diffusion of the inflowing air, whilst the air may also be "conditioned." The defects of these systems very largely arise through the outlet ventilation being controlled more or less directly by natural agencies. The extent to which this weakness becomes marked is governed by the arrangement of the outlet ducts. Where, for example, the outlet duct from a room terminates above a roof, the wind affects its rate of discharge and often upsets the balance of a system. Or again, where a number of outlet ducts terminate in the roof space before discharging into the external air, the difference in the power of their draughts is sometimes so marked as to cause the flow of air to be reversed in one in order to supply a duct with a stronger draught.

A failing common to both vacuum and plenum systems, is their easy disorganization by the opening of doors and windows. Some, to minimize this drawback, advocate the use of locked windows, but the remedy suggested is worse than the disease.

Combined Systems.—An installation that is the least liable to derangement is one in which the plenum and vacuum systems are combined. This combination permits of the control of both the inflowing and outflowing air, and when the propelling and extracting forces are properly balanced, the opening of windows (which is so often desirable) or doors will have no adverse effect upon a system at any other point. The initial and operating costs, however, are necessarily greater in the combined than in the separate systems, on account of the higher initial and operating costs.

Heating Apparatus.—Buildings may be warmed by either of the following:—

(*a*) Open fires.
(*b*) Stoves.
(*c*) Hot-air furnaces.
(*d*) Hot-water apparatus.
(*e*) Steam-heating systems.
(*f*) Gas fires.
(*g*) Electric heaters.

Open Fires.—If viewed only from an economical standpoint, open fires have nothing to recommend them. The heat usefully employed is but a small percentage of that which the fuel yields. All the heat, however, that passes into the chimney is not lost, as a portion is essential to produce the draught. Open fires have a cheerful effect, and it is this property that so strongly appeals to the average Britisher. The chimney also makes a good outlet ventilator. The chief drawbacks of open fires, in addition to wastefulness, are the amount of work they entail, their large share in the pollution of the atmosphere (especially in congested areas), and their over-heating and under-heating effects according to position in room. They are often combined with other forms of heating where economy is of secondary importance.

Hot-Air Furnaces.—In this country, heating by these furnaces finds little favour, although with a well-proportioned and installed system fairly good results may be obtained. Furnace systems are of two kinds, the first where the circulation of the heated air depends upon the force of gravity, the second where the air is propelled or drawn over the heating surfaces by fans, and afterwards forced through the distributing ducts to the apartments to be warmed. The drawbacks associated with furnace heating are often due to the furnaces being too small, to the over-heating of the surfaces, to structural defects, and to the ducts not being properly sized. When fans are employed, the air may be purified and otherwise treated. Furnace systems of heating are less costly than hot-water and steam installations, but they are less durable. Generally speaking, the heated air from furnaces is rather dry.

Hot-Water Gravity Apparatus.—Systems using hot water as the circulating medium may be roughly divided into two classes, (*a*) open systems, and (*b*) sealed systems. Those which are open to the atmosphere are usually termed low-pressure installations irrespective of what the hydrostatic pressure at any point may be.

Sealed systems take two principal forms. In the first, the internal pressure may be raised to any extent desired, this being limited only by the strength of the apparatus. In the second the maximum internal pressure is limited, loaded valves

or mercury seals being used to afford relief. The principal purpose for sealing a system is to raise the boiling-point of the water to a temperature higher than that corresponding with atmospheric pressure.

Low-Pressure Systems.—For dwellings, schools, workrooms, offices, etc., low-pressure hot-water heating with direct surfaces is specially advantageous. The apparatus is easily managed, the temperature of the circulating water can be increased or diminished to suit the weather, the heat emitted may be of a mild nature, all unpleasantness through the over-heating of the dust that accumulates on the radiators may be obviated, the temperature of rooms can be readily controlled, and with ordinary care the apparatus is very durable.

The chief drawbacks associated with low-pressure apparatus are the large heating surfaces and pipes required. Neither do systems which hold large volumes of water, readily lend themselves to automatic regulation.

Sealed Systems.—Small-bore hermetically sealed apparatus can be applied to a variety of uses. At one time this system was largely used for warming dwellings, churches, and other public buildings, but it is better suited for drying-rooms, where high temperatures are necessary, for boiling-pans, for bakers' ovens, and for other industrial uses.

Systems sealed with loading devices, such as valves and mercury seals, are suitable for buildings where for economical reasons moderately high water temperatures are adopted. On account of the higher temperature that can be obtained when compared with open systems, a smaller quantity of heating surface is necessary, smaller pipes may be used, and in consequence the initial cost is less.

Hot-Water Apparatus with Forced and Accelerated Circulation.—As the energy producing the circulation of water in gravity systems is a very limited quantity, the use of these is in consequence restricted. In forced circulating systems, the water is moved by positive means, whilst in accelerated circulating systems the head producing movement is in excess of that in ordinary installations when erected under similar conditions. The principal features of these special systems are: Small pipes can be used in virtue of the quickened movement

of the water; heat can be transmitted long distances without undue cooling of the water; there is greater freedom as regards the way in which the pipes may be arranged; and when installations are of a large size they are less costly to erect.

Steam Heating Apparatus.—These may be divided into the older and the newer forms. The former includes "high" and "low" pressure systems in which the pressure in the return pipes is greater than that of the atmosphere. In the latter, the heating surfaces and the return mains are open to atmospheric pressure. The newer forms of apparatus have been developed at a remarkable pace the last few years, and most of the drawbacks incidental to the earlier forms have been removed.

High-Pressure Steam Heating is usually confined to places where high temperatures are required, such as drying-rooms, stoves, Turkish baths, etc. It is unsuitable for general heating work on account of the highly heated surfaces affecting the quality of the air.

Low-Pressure Steam Apparatus.—Although there is no precise definition of the term "low pressure," it is generally understood to be associated with a system in which the steam pressure is less than 10 lbs. per square inch (gauge pressure). Usually the gauge pressure does not exceed 5 lbs. per square inch.

Low-pressure steam heating is suitable for works, large public and private institutions, and for buildings that are irregularly heated, and where there would be danger of a water system being damaged by frost. Another feature of steam heating is that the boilers readily lend themselves to the automatic control of the rate of combustion.

The principal drawbacks to the early types of apparatus are: The temperature of the heating surfaces cannot be regulated by the valves—they must either be fully on or off; the steam at all times must exceed in temperature 212°, irrespective of external conditions; they are less economical than corresponding systems in which water is used; clicking or hammering sounds are often produced, and a certain percentage of the heating surfaces is ineffective.

Atmospheric Steam-Heating Systems.—These differ princi-

pally from the ordinary low-pressure apparatus in that the returns are open to the atmosphere, and no resistance is offered to the water of condensation other than that due to pipe friction; the steam supply is also restricted to the amount a radiator or other surfaces can condense. The boiler pressure or other source of steam supply is limited to a few ounces per square inch, or to a greater pressure, depending upon the conditions the plant is required to satisfy.

The advantages of atmospheric systems over the earlier forms are: the lower temperature of the heating surfaces, the degree of regulation afforded by the radiator valves, and the units into which a plant can be divided with a simple arrangement of piping. Their operating costs are also less than those of ordinary steam systems. Defective appliances often cause waste of steam, but this can be avoided by the adoption of suitable fittings, and the proper regulation of the steam supply.

Vacuum and Vacuo Vapour Systems of Steam Heating.—These take many forms, differing principally in their mode of operation and in the appliances that are used. Speaking broadly, they differ chiefly from atmospheric systems in the manner the differential pressure is produced to cause the circulation of the steam, and to remove the air from the heating surfaces. In vacuum systems some form of exhauster is employed, whilst in atmospheric systems the differential pressure is due to the direct fall of pressure upon the steam entering the heating surfaces. The principal vacuum systems may be divided into two classes: the first in which an independent air line is employed for the removal of the air, and the second, that in which the air and water of condensation are conveyed by the same pipe.

A good vacuum system has a number of points in its favour, such as where the condensation is unable to gravitate to the point desired, and where exhaust steam is the heating medium. They may be designed to be very flexible in operation, and the heating surfaces may be maintained at a high or comparatively low temperature, according to the intensity of the heat desired. For example, assuming that it is found economical to operate a system under 15 inches of vacuum, it should be possible during the mild weather, by restricting the supply, to circulate the

steam at a temperature of about 180° Fahr., whilst with lower external temperatures, the steam may be increased until eventually its temperature in the heating surfaces coincides with the initial pressure.

All so-called vacuum systems, however, are not a success. For example, in a certain class of apparatus, where the vacuum generated depends largely on the curtailment of the steam supply, trouble may arise through imperfect drainage. In other cases, the cost of creating the vacuum is out of all proportion to the benefits that are obtained.

The efficiency of a vacuum system is chiefly dependent upon the form it takes, and upon the kind and quality of the fittings that are used.

Heating by Gas.—Of late the advantages of gas fires for heating purposes have been greatly discussed, but as a general means for warming buildings gas apparatus will take a different and more economical form. Gas fires, however, are convenient for heating rooms that are temporarily used, especially in mild climates, for they are clean and easily put into and out of use. The chief drawbacks associated with gas fires occur when they are of defective design. To be satisfactory, practically the whole of the heat they transmit should be from the radiant used. The design should prevent the exterior surfaces being raised to high temperatures, and ample provision should be made for carrying off the products of combustion.

Gas heaters that warm chiefly by convected heat, and those that permit the products of combustion to escape into the surrounding air, have nothing from a health standpoint to recommend them.

The smoke nuisance or aërial pollution that exists in industrial and densely populated centres cannot much longer continue, and so far as the domestic side of the problem is concerned, the trouble can be greatly minimized by the use of gaseous fuel for small hot-water and steam heating apparatus. Before this method of heating can be extensively adopted, gas, however, will require to be sold at a much cheaper rate. The latter aspect of the problem should not be a difficult one, especially in view of the lower qualities that can be used.

Another use for gas that is becoming more common is the

heating of water for domestic and other purposes, and its further adoption will doubtless increase as its value in this direction is realized.

Heating by Electricity.—This form of energy is a very convenient one for heating purposes, but it is only in very special cases that it can be economically employed. Where electricity for its production is dependent upon fuel, it is placed at a disadvantage so far as heating is concerned, as only a very small percentage of the total heat energy of the fuel can be transformed into electrical energy.

For popularizing electricity for heating rooms, some municipalities sell it at a specially low rate, and undoubtedly more could be done in this direction. This is especially the case where the period of minimum load at a generating station coincides with the time the heat is principally wanted.

District and Large Central Heating Plants.—Reference has been made already to the smoke-polluted atmosphere of towns and the large centres of industry. In the case of factories and large public or private institutions, the acute trouble arising from the production of smoke can be remedied by the introduction of appliances that better regulate the combustion of fuel, by improved types of furnaces, and by the substitution of electrical power, oil and gaseous fuels for the coal consumed in small power plants.

On the domestic side, a good method of reducing the production of smoke is by district and large central heating systems in which either water or steam is circulated through the pipes. Such plants have also advantages over small independent systems in that the labour for attention is centralized, whilst a householder can procure heat at any time without any inconvenience.

To what extent district heating is practicable where the heating medium is conveyed through long underground pipes, depends principally upon the demand for this source of heat, the climate, and the length of the heating season. In America, district heating has been extensively adopted, but the same favourable results could not be obtained in Great Britain where underground pipes are of considerable length owing to its much milder climate. There are, however, numerous localized areas

in all our towns, more or less congested, that lend themselves admirably to isolated heating plants, whilst in other cases single groups of buildings could be economically heated from one central source.

District Heating can be carried out to greater advantage where the same station supplies both heat and electrical energy. The units of combined plants may be so arranged that the exhaust steam forms a large portion of the heating medium, and probably the best results would be obtained when the volume of the "exhaust" was just sufficient to supply the demand for heat during the average winter temperature. For lower external temperature the deficiency can be made good by the addition of "live" steam, whilst for milder conditions the surplus steam may be passed to exhaust, or sent to the condenser.

District systems take two principal forms. In the first, either "live" or "exhaust" steam or a mixture of both is delivered from the central station to the heating surfaces. The water of condensation is either passed to waste or returned to the source of heat, the former being the more common practice. In the other system, water is circulated between the station and the heating surfaces, this in turn receiving its heat from either "live" or "exhaust" steam or from both.

As regards the relative merits of hot water and steam systems for district heating, each has its own special points, and the choice depends upon the chief conditions to be met.

Generally speaking, steam heating is advantageous for industrial centres in that the steam can be utilized for a variety of purposes in addition to general heating work.

Hot-water systems are largely used for residential districts. The water temperature can be varied at the station to suit the changing atmospheric conditions, and when condensing engines are used, this may be done by regulating the degree of vacuum at the condenser. Other features of water systems are, that the transmission losses are considerably less than those of steam plants, and less back-pressure is put upon the engines.

CHAPTER II

HOT-WATER CIRCULATION

THERE are three phases of circulation applicable to hot-water apparatus: (*a*) natural or gravity; (*b*) accelerated; and (*c*) forced.

Gravity Circulation depends primarily upon the application of heat at a low point of an apparatus, when the pressures exerted by the flow and return columns are rendered unequal.

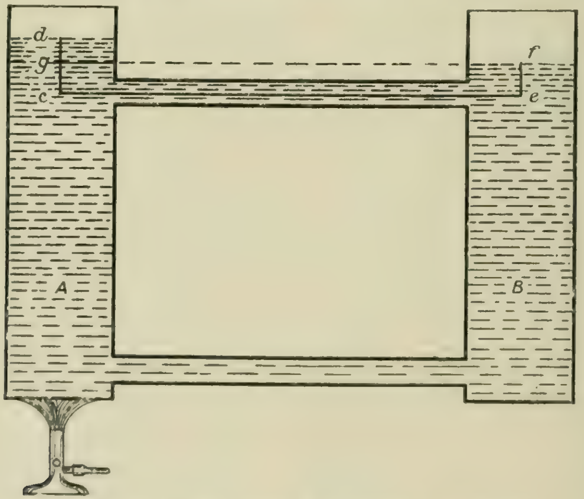


FIG. 1.—Tredgold's method of explaining circulation.

Although this explains the cause of gravity circulation in a general way, many do not appear to grasp the fact.

Probably Mr. Tredgold was the first who attempted to account for the circulation of water in a heating system, whilst Hood, in a later work, takes exception to his conclusions, and

endeavours to show his reasoning is not correct. The explanations of circulation by these gentlemen are interesting and instructive in showing the points on which they differ.

Tredgold's account of circulation is as follows: "If the vessels A and B (Fig. 1) and the pipes connecting them be filled with water, and heat be applied to A, the effect of heat will expand the water in the vessel A, and the surface will in consequence rise to a higher level *d*, the common level being *gf*. The density of the fluid in the vessel A will also decrease in consequence of its expansion, but as soon as the column

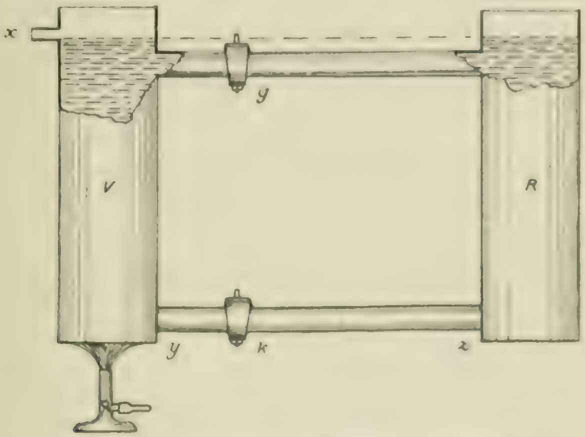


FIG. 2.—Hood's method of explaining circulation.

ed (above the centre of the upper pipe) is of greater weight than the column *fe*, motion will commence along the upper pipe from A to B, and the change this motion produces in the equilibrium of the fluid will cause a corresponding motion in the lower horizontal pipe B to A."

Hood makes the following statement: "Suppose the apparatus (Fig. 2) to be filled with cold water, and the two stop-cocks are closed. On applying heat to the vessel V, the water it contains will expand in bulk, and a part of it will flow through the waste pipe *x*, which is so placed as to prevent the water rising higher in the vessel V than in that of vessel R. The water which remains in the vessel V after it has been heated will evidently be lighter than it was before owing

to a portion having passed through the waste pipe x, although its height will remain unaltered. Suppose now the two cocks, g and k, to be simultaneously opened: the hot water in the vessel V will immediately flow towards R through the upper pipe, and the cold water in R will flow to V through the lower horizontal pipe, although by Tredgold's hypothesis, unless

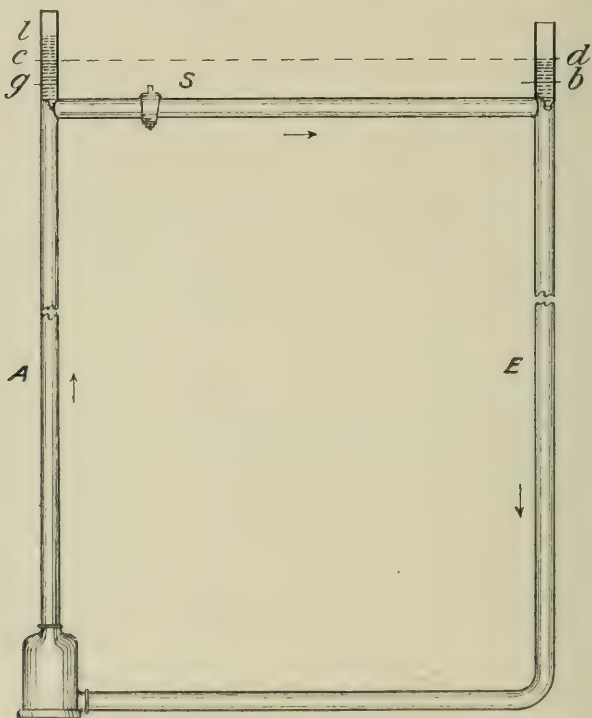


FIG. 3.—Illustrating cause of circulation.

the water in vessel V rose to a higher level than that in the vessel R no circulation could take place." Mr. Hood continues: "Assume heat to be applied to the vessel V (Fig. 2): the heated particles rise through the colder ones, which sink to the bottom by their greater specific gravity, and they in turn become heated and expand like the others. As soon as the water in vessel V begins to acquire heat and to become

lighter than that in vessel R, the water in the lower horizontal pipe is pressed by a greater weight at Z than at y, and it therefore moves towards V with a velocity and force equal to the difference in pressure at the two points y and Z. The water in the upper vessel R would now assume a lower level were it not that the upper horizontal pipe furnishes a fresh supply from V to replenish the deficiency."

Upon the first perusal Tredgold's explanation may not be very clear, but briefly expressed it is: that the circulation of water is due to the equilibrium of the "flow" and "return" columns being destroyed by the overflow of the expanded water as represented by gl in Fig. 1. Hood's views stated briefly are: that circulation is due to the greater density of the return water, and that the increment of expansion has nothing to do with it.

The reasons given by Tredgold and partly those by Hood are correct so far as they are carried, but Hood errs when he tries to prove that Tredgold is wrong. The overflow r in Hood's apparatus (Fig. 2) does not dispose of the increment of expansion when the stop-cock k is opened, and his view has been simply narrowed through the use of the additional cock q.

The following may aid in the elucidation of the point under consideration. Assume an apparatus with a circuit of indefinite height, as in Fig. 3, to be filled with water to the level gh, with the stop-cock s closed. If heat be applied to the boiler the water in the flow pipe may be expanded so as to reach the point l. So long as the stop-cock is closed, the pressures in columns A and E are in equilibrium, but no circulation occurs in the return pipe although the two columns differ in density and are in direct communication by means of the lower horizontal pipe. If, now, the stop-cock s be opened, the equilibrium is at once destroyed, and circulation begins owing to the increment gl being able to flow towards the return. Now, as the total pressure in column A has been diminished by an amount, say, el, and the pressure in column E increased by the amount hd, the total head causing circulation is $el + hd$, the sum of which is equal to the increment of expansion gl.

If an apparatus takes a form similar to that of Fig. 4, the effect is the same although the piping differs somewhat from

Fig. 3. For example, if *nm* represents the increment of expansion when the return is temporarily disconnected from the flow, it is apparent that it will produce two unequal forces when the passage through the upper horizontal pipe is clear.

Circulating Head, or the power that causes the movement

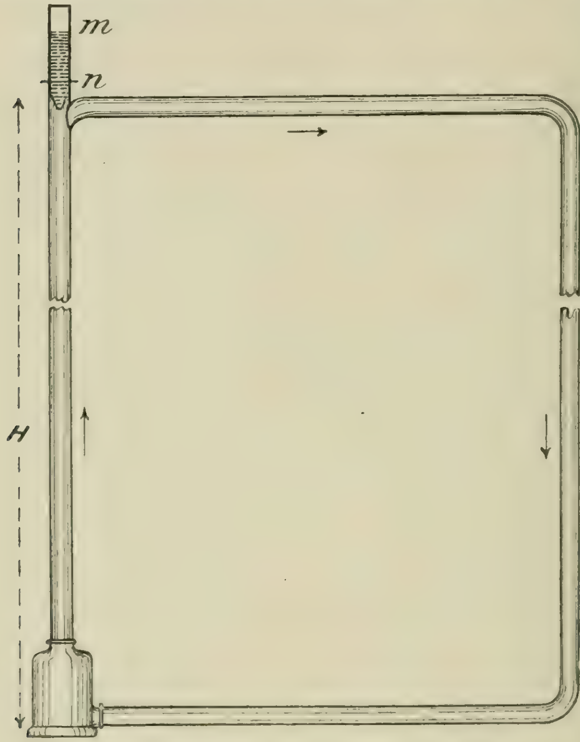


FIG. 4.—Illustrating the increment of expansion.

of the water, is the same thing as the increment of expansion, and is represented by *gl* in Fig. 3 and *mn* in Fig. 4. It may be calculated by the formula—

$$h = 12H\left(\frac{w}{w_1} - 1\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where h = circulating head in inches,
H = height of circuit in feet,
w = average density of water per cubic foot in return,
w_1 = average density of water per cubic foot in flow.

Circuit Height.—The *height* of a circuit in contradistinction to the *circulating head* is the vertical distance between the highest and lowest points where the water circulates. In the case of main circuits which do not dip beneath a boiler, the fire bars are taken as the lowest point, although some assume it as the centre of the fire pot.

Example 1.—The height of a circuit is 20 feet, and the average temperatures of the flow and return water 160° and 130° F. Determine the value of the circulating head.

By formula 1 $$h = 12\text{H}\left(\frac{w}{w_1} - 1\right).$$

In the Appendix, p. 312, the weight of water for 160° and 130° F. is given as 60·99 lb. and 61·56 lb. per cubic foot respectively.

Substituting these values—

$$h = 12 \times 20\left(\frac{61 \cdot 56}{60 \cdot 99} - 1\right)$$

when $h = 2 \cdot 241$, or, say, $2\frac{1}{4}$ inches.

Velocity of Gravity Circulation.—If friction were absent, the velocity of circulation would be nearly equal to the velocity attained by a body when freely falling through a height equal to the *circulating head*. Pipe friction, however, is a more or less considerable quantity, and therefore requires to be taken into account.

The following general formula may be used for determining the velocity of circulation—

$$\text{V} = c\sqrt{\frac{\text{H}d\left(\frac{w}{w_1} - 1\right)}{l}} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where V = velocity in feet per minute.
" c = a coefficient that varies with the diameter.
" H = effective circuit height.
" d = diameter of circuit in inches.
" l = length of circuit in feet.
" w and w_1 = the densities of the return and flow waters per cubic foot.

APPROXIMATE VALUES OF c FOR SIMPLE CIRCUITS.

Circuits of	1 to $1\frac{1}{2}$-inch	diameter	$c = 600$
,,	2 to 3 ,,	,,	$c = 700$
,,	$3\frac{1}{2}$ to 5 ,,	,,	$c = 750$

Example 2.—Determine the velocity of circulation of a 3-inch diameter circuit that takes the form of Fig. 4. Let its length be assumed as 360 feet, its effective height 20 feet, and the average temperatures of the flow and return waters 155° and 145° F. respectively.

By formula 2—

$$V = c\sqrt{\frac{Hd\left(\frac{w}{w_1} - 1\right)}{l}}$$

For 145° F. $w_1 = 61{\cdot}29$ lb.; for 155° F. $w = 61{\cdot}1$ lb., and c is given for a 3-inch diameter pipe as 700.

$$\text{Substituting values, } V = 700\sqrt{\frac{20 \times 3 \times \left(\frac{61{\cdot}29}{61{\cdot}1} - 1\right)}{360}}$$

$$V = 700 \times 0{\cdot}0227$$

$$\text{when } V = 15{\cdot}9 \text{ feet per minute.}$$

This calculation shows what a feeble current is obtained under the conditions given, although the circuit is of moderate height.

Dipped or Trapped Circuits.—It is sometimes necessary to dip or trap a circuit, and although, as a general rule, this should be avoided as far as it is practicable, yet dips can be introduced in many cases without greatly impeding the circulation. At all dips, vent pipes should be provided, for it is imperative that the air should escape freely when a system is charged with water, or when it is in operation.

The effect of dips is to reduce the circulating head, and the velocity of flow is directly proportional to the square root of that factor when the other conditions remain unaltered. The circulating head of trapped circuits may be estimated by formula 1 after first determining the average density of the ascending and descending columns of water.

Figs. 5 and 6 give two trapped circuits where the rate of cooling is assumed to be as shown. For convenience, the densities of the water for the temperatures under consideration

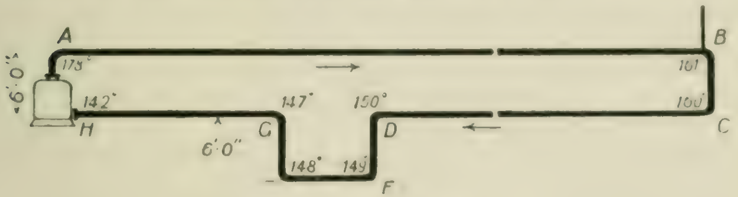


Fig. 5.—Trapped circuit.

Fig. 6.—Trapped circuit.

are here given, whilst the method of ascertaining the average densities per cubic foot of the ascending and descending columns of water is as follows:—

Temp. Deg. F.	Weight lb. per cubic ft.	Temp. Deg. F.	Weight lb. per cubic ft.	Temp. Deg. F.	Weight lb. per cubic ft.
142	61·34	148	61·22	160	60·98
143	61·32	149	61·20	161	60·96
147	61·24	150	61·18	178	60·59

Particulars of Fig. 5.

Ascending columns.			Descending columns.		
Column.	Average temp.	Density per cubic ft. in lb.	Column.	Average temp.	Density per cubic ft. in lb.
HA	178°	60·59	BC	160°	60·98
FG	148°	61·22	DE	150°	61·18
		2)121·81			2)122·16
Average density =		60·905	Average density =		61·08

This simple method of ascertaining the average density is only applicable when the total height of the ascending columns is equally divided between AH and FG, and that of the descending columns between BC and DE, as in Fig. 5. Where the parts forming the columns are unequal, as in the ascending ones of Fig. 6, the procedure is rather different.

PARTICULARS OF FIG. 6.
Ascending columns.

Column.	Average temp. deg. F	Density per cubic ft	×	Height of column ft.	=	Foot-lb.
SL	178	60·59	×	6	=	363·54
PR	147	61·24	×	2	=	122·48
						8)486·02
		Average density =				60·752

The average temperature for the descending column MO may be taken at 160°, and the density corresponding with this is 60·98 lb. per cubic foot.

Velocity of Circulation through Trapped Circuits.—The velocity in trapped circuits may be ascertained approximately by letting w and w_1 in formula 2 represent the average densities of the descending and ascending water columns.

Example 3.—Determine the probable rate of flow through a circuit that takes the form of Fig. 5, the height being 12 feet, length 220 feet, and of 2-inch bore. Assume the water to cool at the rate given.

By formula 2—

$$V = c\sqrt{\frac{Hd\left(\frac{w}{w_1} - 1\right)}{l}}$$

On p. 23 the average density of the ascending columns was found to be 60·905 lb., and that of the descending columns 61·08 lb. per cubic foot. The value of $c = 700$.

Substituting values—

$$V = 700\sqrt{\frac{12 \times 2 \times \left(\frac{61 \cdot 08}{60 \cdot 905} - 1\right)}{220}}$$

$$V = 700 \times 0 \cdot 017,$$

when V = 11·9 feet per minute.

Example 4.—Assuming the length and diameter of circuit in Fig. 6 are the same as in the previous example, what will be the probable velocity of circulation if the water cools as shown?

By formula 2—

$$V = c\sqrt{\frac{Hd\left(\frac{w}{w_1} - 1\right)}{l}}$$

The average density of the ascending columns was found on p. 24 to be 60·752 lb. per cubic foot, whilst that of the descending column is 60·98 lb.

Substituting values—

$$V = 700\sqrt{\frac{8 \times 2 \times \left(\frac{60 \cdot 98}{60 \cdot 752} - 1\right)}{220}}$$

$$V = 700 \times 0{\cdot}016,$$

$$\text{when } V = 11{\cdot}2 \text{ feet per minute.}$$

To show the application of formula 2 to the problems indicated by Figs. 5 and 6, the rate of cooling has been considered to have been equal over the whole of the circuits. In practice, however, this effect would not be realized, for more heating surface is usually concentrated at one point than at another; moreover, the cooling effect is greatest where the water is hottest. These factors therefore require to be taken into account, for a marked cooling of the water at a certain point will either impede or accelerate circulation. Were radiators connected to the circuit between the points DC, Fig. 5, the water may be cooled to a greater extent than shown, but the rate of circulation would be little affected. On the other hand, if the water in the pipe OP, Fig. 6, were cooled more than shown, this would adversely affect the circulation, owing to the greater resistance offered by the column PR.

Irregular or Spasmodic Circulation.—The principal causes for irregular motion in a heating system are due to the accumulation of air and to trapped circuits. The remedy for the first is simple, for all that is required is properly pitched pipes, and a well-vented system. As a rule, the pitch of

pipes should not be less than 1 inch in 10 feet, but should be greater for those of small-bore.

As already shown the adverse effect of dips is when they tend to equalize the pressure of the ascending and descending water columns. If this condition is realized, circulation ceases, and equilibrium is only destroyed either by the overheating at the boiler on the one hand, or by the abnormal cooling of some portion of the return water on the other. In badly designed systems, irregular circulation often occurs in branched circuits, and especially in those of small bore.

Forced Circulation.—A single circuit with forced circulation is given in Fig. 7. This form of circulation is especially suitable where a series of detached buildings is to be warmed from a

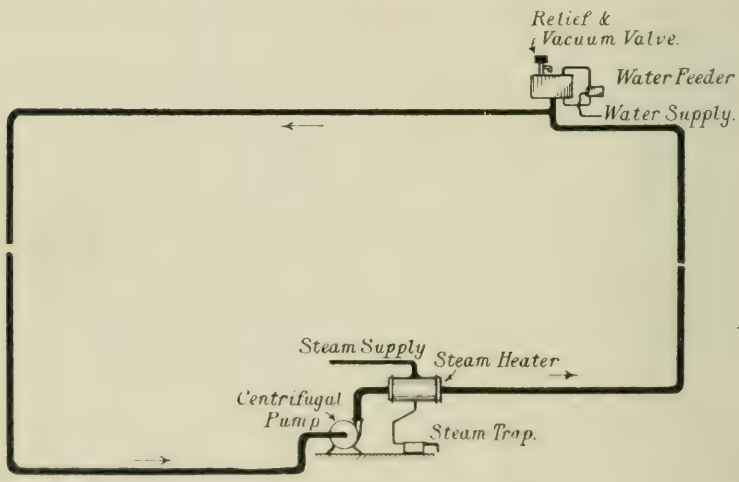


FIG. 7.—Forced circulating system.

central source; also for large buildings and structures where gravity circulation is unreliable or otherwise unsuitable. With forced circulation, the pipes may be run and fixed in any position desired, irrespective of dips. The greatest drawbacks associated with it are the cost of operation and the greater attention required, when compared with gravity circulation.

The speed of circulation should be governed by the extent of the resistance offered by the piping and appliances used,

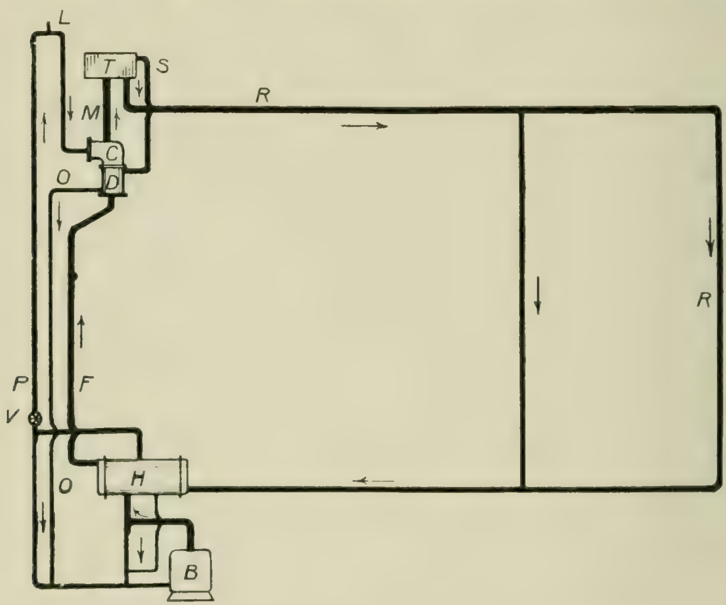
but the "critical" velocity may lie anywhere between 2 feet and 8 feet per second. The critical velocity is the one that gives the most economical results with reference to the cost of operation and to the initial outlay as a whole. In comparatively small plants, the critical velocity is not so important, and in these, the circulating speed is often fixed with reference to a given size of motor, and to the efficiency of the pump that it is proposed to use.

The power for circulating water by means of a pump varies roughly with the cube of the speed, whilst the amount of heating surface a given size of pipe will serve, varies directly with the velocity of circulation, provided the temperature drop remains unaltered. Assume that water is moved through a circuit at the rate of 100 feet per minute in order to serve 500 square feet of heating surfaces, and one unit of power is absorbed. If now the velocity is raised to 400 feet per minute, this will supply four times the amount of heating surface, or $\frac{400 \times 500}{100} = 2000$ square feet. The energy, however, to produce this rate of flow would require $\frac{1 \times 400^3}{100^3} = 64$ units, or sixty-four times the original quantity. Thus it becomes clear that the saving effected in the initial cost of an apparatus by adopting very high circulating speeds may be offset by the greater operating charges.

Accelerated Circulation.—There are various ways in which the circulation of water may be accelerated, such as by providing apparatus whereby the temperature of the water is raised above the normal amount, by the injection of steam or air into the flow pipe of a system, by producing a partial vacuum in a circulating tank through the condensation of steam, or by mechanically increasing the "circulating head." The general advantages derived from accelerated circulation are similar to those of forced circulation, but the force producing motion is not of so "positive" a nature, and hence the speed that can be attained is limited. Each particular method of acceleration has its own special merits and limitations, so that what may give the most satisfactory results in one case may be disadvantageous in another.

Although the same precautions with respect to the gradients of pipes need not be observed as in gravity systems, yet it is desirable to do so, for this permits accelerating apparatus to be operated by gravity circulation during the milder portions of a heating season.

A method of accelerating circulation by means of steam is shown in Fig. 8. It represents the principle introduced by



L = automatic vent.
T = expansion tank.
S = overflow and exhaust pipe.
R = return from tank T.
M = mixing pipe.
C = circulation.
D = condenser.
O = condensation return.
P = steam supply.
V = steam valve.
F = pipe to circulator.
H = reheater.
B = steam boiler.

FIG. 8.—"Reck System." *

Captain Reck, and is called the "Reck System." Steam is generated in the boiler B, from which it flows to the re-heater H, which takes the form of a straight-tubed calorifier with water-way ends. The steam that is not condensed in the re-heater H flows through the pipe P to the top of the apparatus, the pipe

* The reheater H is frequently omitted.

being returned to join at C, where the steam mixes with the water circulating through the flow-pipe F. The mixing of steam and water at C has the effect of substantially diminishing the pressure of the ascending columns when compared with that of the descending ones. The pipe M is directly joined to the expansion tank T, the latter providing space for the expanding water, and acting as a barrier to prevent the passage of steam into the water circuit. From the tank T, two pipes are taken; that indicated by R completes the circuit for the water by joining with the re-heater H, whilst the pipe S serves as the overflow to return surplus water to the boiler, and also to convey any steam to the condenser D.

For any given installation, the rate of acceleration largely depends upon the height of that portion of the flow pipe M which represents the vertical distance between the circulator C and the expansion tank; the greater this height, other conditions being equal, the more rapid the circulation. As a rule, however, the vertical distance between the circulator and expansion tank (even in large plants) is limited to a few feet, for this also determines the steam pressure at which the boiler will require to be operated. Assuming, for example, that steam is not generated, or that it is cut off from the circulator by the valve V, it is obvious that the water will rise in the dipped portion of the pipe P to the level of that in the expansion tank. The pressure, therefore, must be adequate to dislodge this water before the steam can enter the circulator. It is essential to carry the steam pipe higher than the expansion tank in order to avoid the flooding of the boiler, whilst to prevent the water being siphoned from the expansion tank through a partial vacuum forming in the steam pipe, an automatic air valve is provided at L.

To permit of the escape of air from the apparatus shown by Fig. 8, an automatic relief valve is usually attached to the condensers, or an open pipe may be provided immediately beneath so long as some form of mechanism is employed to control the steam automatically to the circulator. The upper portion of pipe R may also be vented to the atmosphere if desired. When steam is introduced into a body of water some means should be introduced to obviate hammering or cracking

sounds. In the "Reck" apparatus this is done by the use of fine-mesh gauze, which has the effect of breaking up the steam into a very fine spray. Special forms of injectors may also be used for the same purpose.

The rate of circulation in a system resembling Fig. 8, may be ascertained by formula 2, provided that the condenser properly performs its function.

Example 5.—Determine the velocity of circulation when the height of the column F in Fig. 8 is 35 feet, and contains water with an average temperature of 170° F., column M 5 feet high, with a temperature of 212° F., whilst free steam occupies one-fifth the pipe area. Take the average temperature of the water in R as 180° F., the length of the circuit 300 feet, and its diameter 3 inches.

The density of the water at 212° F. is 59·76 lb. per cubic foot, whilst that of the steam may be taken as 0·038 lb. per cubic foot. From these values the density of the mixture in pipe M will be $\frac{59{\cdot}76 \times 4}{5} + \frac{0{\cdot}038}{5} = 47{\cdot}81$ lb. per cubic foot. The average density of the ascending columns may be obtained by the same procedure as in trapped circuits.

Column.	Temp. deg. F.	Density per cubic ft. × Height of column in ft. = Foot-lb.
F.	170	60·78 × 35 = 2127·30
M.	212	47·81 × 5 = 239·05

40)2366·35

Average density per cubic foot of ascending column = 59·16 lb.

The average density of the water of the descending column R for 180° F. will be 60·55 lb. per cubic foot.

By formula 2—

$$V = c\sqrt{\frac{dH\left(\frac{w}{w_1} - 1\right)}{l}}$$

For a 3-inch diameter circuit $c = 700$, and its height is 40 feet.

Substituting values—

$$V = 700\sqrt{\frac{3 \times 40 \times \left(\frac{60 \cdot 5}{59 \cdot 16} - 1\right)}{300}}$$

$$V = 700 \times 0 \cdot 097,$$

when $V = 68$ feet per minute.

Suppose now that the circulator C is thrown out of use by closing the valve V, and the average temperatures of the water in F and R are 170° and 150° F. respectively. Where all the

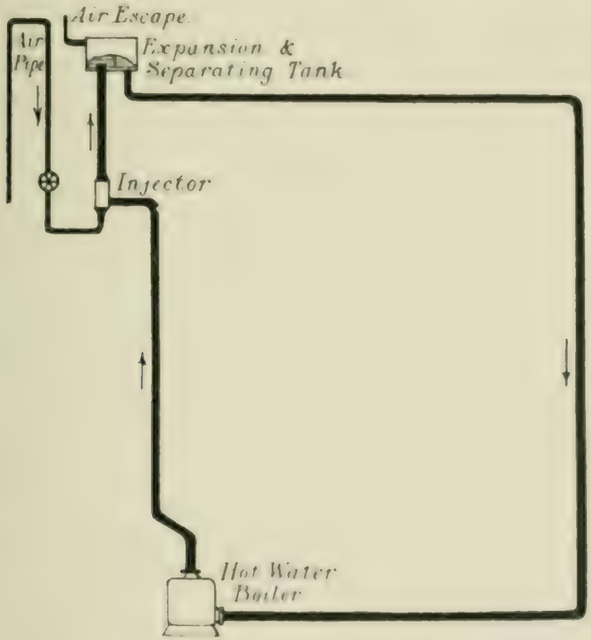


Fig. 9.—Accelerated circulating system.

other conditions remain unaltered, the velocity will be directly proportional to $\sqrt{\frac{w}{w_1} - 1}$. For the latter case, therefore, the rate of circulation would be

$$V = 68\sqrt{\frac{\frac{61 \cdot 2}{60 \cdot 78} - 1}{\frac{60 \cdot 55}{59 \cdot 16} - 1}}$$

when $V_i = 36 \cdot 72$ feet per minute.

Thus, under gravity for the conditions given the speed of circulation is diminished by 46 per cent.

If a supply of compressed air is available the circulation could be accelerated by an injector as in Fig. 9. Air, however, is not so satisfactory or economical as steam for the accelerating agent.

CHAPTER III

SYSTEMS OF PIPING FOR HOT-WATER GRAVITY APPARATUS

THE most suitable method of piping for a particular case will depend very much upon the size of the installation, upon the location of the heating surfaces with respect to one another, upon the facilities for pipe runs, and upon the resistance offered to the circulation of the water.

There are two distinct methods of piping, these being designated as "one-pipe" and "two-pipe" systems, and each may be arranged on either the up-feed or down-feed principle. It is not always desirable to adhere rigidly to any one piping system, but the circumstances of a case may be better met at times by a modification or combination of these systems.

One-Pipe Up-feed System. This is the simplest form of piping, and single circuits are of equal bore for their whole length. The "one-pipe" system is advantageous, in that short circuiting cannot take place, and this makes it especially suitable where the dipping or trapping of circuits is unavoidable. As a general rule, this system is better for supplying comparatively small amounts of heating surfaces than for large ones owing to the cooling effect through the "flow" and "return" water being accommodated in the same pipe. A very pronounced temperature drop can always be avoided by having the pipes of relatively large bore, but this may be undesirable unless another form of piping is unsuitable.

Fig. 10 gives a single main circuit. It is well to keep the horizontal portions of the risers as short as practicable, although there is no serious objection to their being rather prolonged as at A and B, provided the pipes are of suitable size and are given a good pitch. When joining the risers or branches to a

circuit it is desirable that the "flows" be taken from the top, whilst the returns re-enter at the side; this arrangement aids the circulation through the "branch" circuits, besides tending to separate to some extent, the cooled from the hotter water. In Fig. 10 the highest part of the main circuit is shown at H,

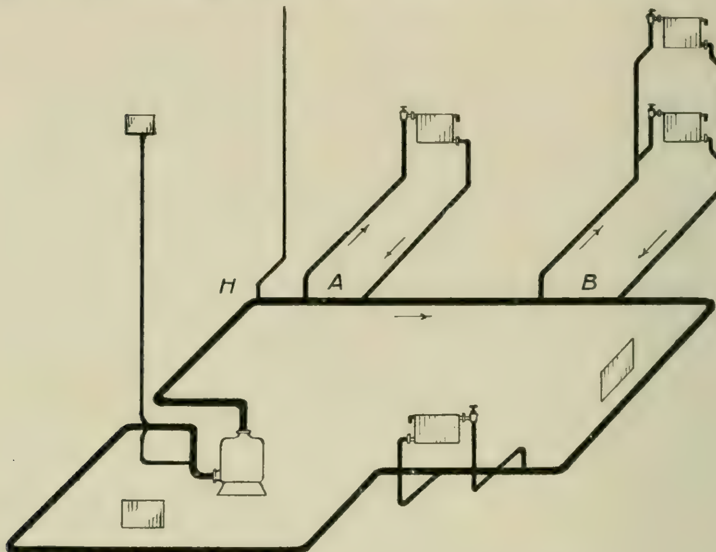


FIG. 10.—One-pipe system: Single main circuit.

where an air pipe is provided; from that point the circuit should have a downward pitch to where the boiler is located, and then be dropped to join with the lower connection.

Although in Fig. 10, H is made the highest part of the main circuit, this need not necessarily be so, but it is the better practice in a "one-pipe" system to make the highest point as near to the boiler as possible. In other words, it is advantageous to limit the length of the "flow" in order that the cooling water can gravitate directly to the source of heat.

In Fig. 11 a divided circuit is given, and this may with advantage be adopted in many cases. With divided circuits, however, or where a loop is introduced as a minor circuit, care should be exercised in making the return connections. Before joining with one common "return" it is a good plan to

drop all the branches, as this causes the water to circulate in the right direction and also aids its movements. With the exception of the short pipe D, the remaining portion of the

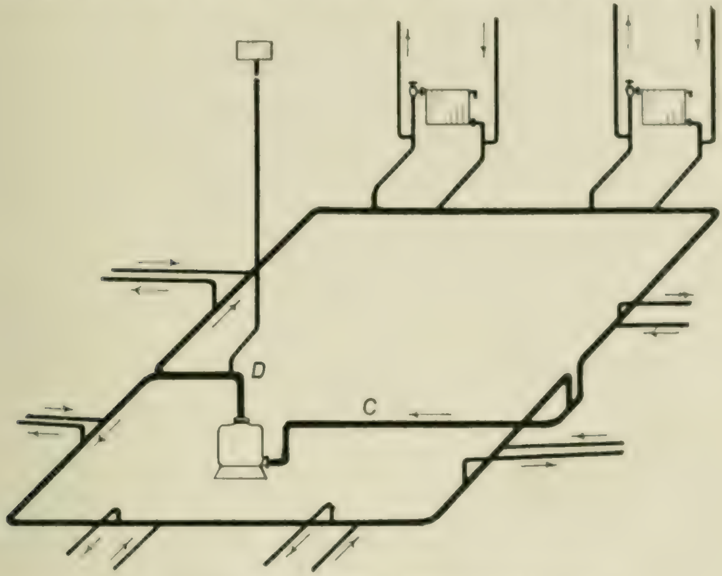


FIG. 11.—One-pipe system: Main circuit divided.

circuits acts as "returns," the air from the heated water being principally relieved by the expansion pipe. Pipes D and C should, of course, be larger than the others.

Fig. 12 gives a "one-pipe" system, where it is essential to dip the circuit beneath a number of doorways. A case of this kind sometimes occurs where corridors are heated by radiators, or where a heating apparatus is erected after a building is completed. The object of carrying the flow pipe above the top floor is to increase the circulating head, but in order for this to be attained, a temperature difference must occur in the upper ascending and descending columns. For a case like Fig. 12, an advantage would be gained if the distance between the pipes at *x* were considerably greater. The air pipe provided at *a* is principally of use when charging the apparatus. From the horizontal pipes, the air would tend to gather in the radiators, from which it could be discharged by opening the air valves.

Another "one-pipe" system is illustrated by Fig. 13, where the heating surfaces on two floors are served by a single circuit.

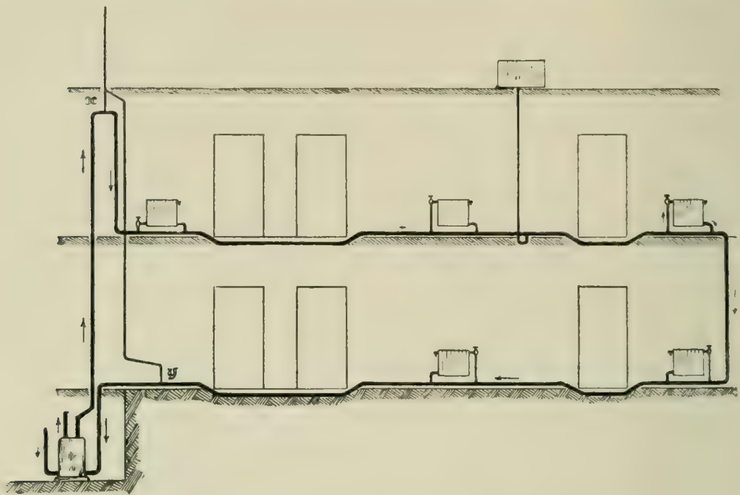


FIG. 12.—One-pipe system where circuit dips beneath doors.

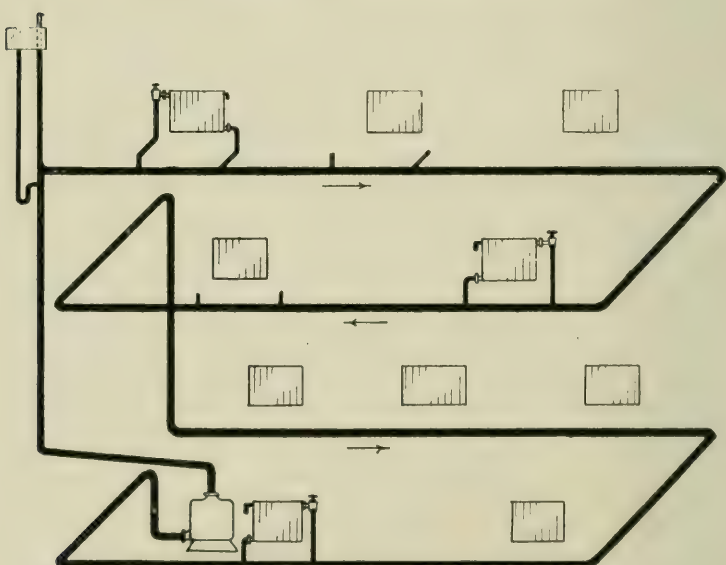


FIG. 13.—One-pipe system.

This is not an economical arrangement as regards the piping, and, unless of a large bore, the temperature difference between the first and last radiator would be pronounced. Instead of a large single circuit being employed for the case shown, it is better to provide an independent circuit for each floor, this being more economical, and also affording a greater degree of flexibility in operation. With separate circuits, stop valves could

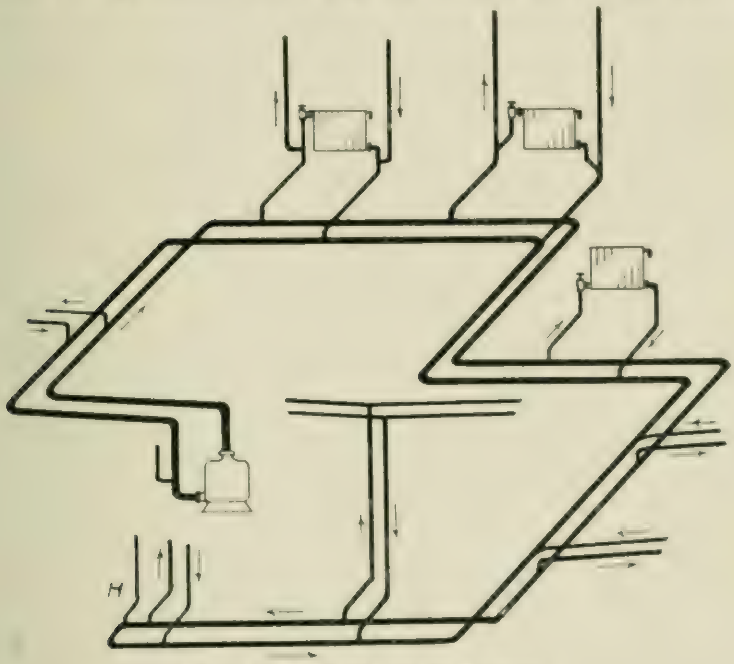


FIG. 14.—Two pipe "up-feed" system.

be provided on the return pipes, and either the one or the other could be partially cut out of use when desired. Should stop valves, however, be fixed upon both flow and return pipes, a safety valve, or relief pipe, should be provided.

Two-Pipe System.—This is the oldest method of piping, and differs from the "one-pipe" or circuit system in that the various flow and return connections join with separate pipes. The main circuit is graded, whilst that of a simple "one-pipe" system is of uniform bore.

A "two-pipe" system is specially suited for buildings where long circuits are required, and owing to the returns from the heated surfaces joining with the main returns, there is a nearer approach to uniformity of temperature in the first and last radiator on a circuit.

To be a success, a "two-pipe" system must be well designed, or some portion of it may not be heated properly through the short circuiting of the water. For this reason, the dipping or trapping of pipes should be avoided, as the water will necessarily circulate the more freely along the path that offers the least resistance.

A general arrangement of the piping is shown in Fig. 14. Here the highest point in the main circuit is at the centre H. As a rule, the radiators on the lower flats are the slowest to get hot, but by joining the risers, as at the upper left radiator, and properly sizing them, the heating up of the radiators on the different floors is more nearly equalized. The arrangement on the right tends to favour the heating surfaces on the higher floors, for a direct and easy passage is provided to those points. Only one main circuit is shown, although for many buildings it would be desirable or necessary to introduce two or more. The number of units into which a system should be divided will largely depend upon its size, the location of the boiler with respect to the heating surfaces, and the degree of regulation desired. Speaking from an economical standpoint, two or more main circuits will have an advantage over one in those cases where a boiler is centrally placed. On the other hand, if a boiler is located near to one end of a building, the cost of the piping begins to rise as the number of separate units is increased.

Pitch of Pipes.—For the main circuits of "one" or "two-pipe" systems, the horizontal piping should have a pitch of from 1 inch in 10 feet to 1 inch in 20 feet. The quicker the pitch the better. As regards the horizontal portions of risers, these should be given, where possible, a pitch of about 1 inch per foot.

Drop, Overhead, or Down-feed Systems are represented by Fig. 15. This method of piping is very suitable for high buildings, where the heating surfaces on the different floors can be located immediately above one another. From the boiler the

flow pipe is carried as directly as possible to the highest part of a structure, whilst the radiators are supplied by the vertical pipes. The whole of the piping from the point P is arranged to take the form of returns. For some structures, this method of piping is not suitable, but where it can be adopted it is the most economical one. The principal advantages of "drop" systems arise through the greater freedom of circulation, and the smaller pipes that can be used, and to the fact that air valves on the heating surfaces are not required.

In Fig. 15, two arrangements of drop pipes are shown.

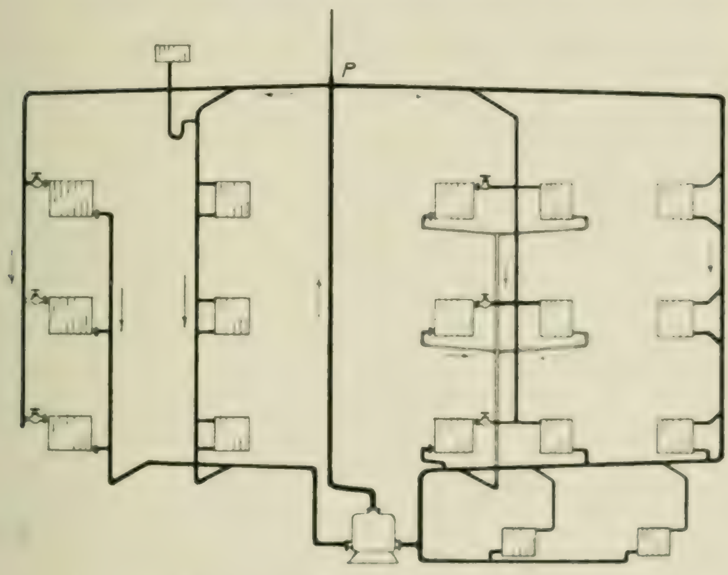


FIG. 15.—Overhead or down-feed system.

The use of single drop pipes necessitates their being of equal bore from end to end, whilst if they take the double form they may be varied in size. The upper horizontal distributing, and the lower intercepting pipes are graded as well, and they may be arranged in various ways, according to the circumstances of the case in hand. To the right of Fig. 15 two radiators are shown on the lowest floor, these being supplied with separate branches instead of continuing the principal drop

pipes to the lowest points. There is much freedom with regard to the choice of details in overhead systems, but every care should be observed to prevent air being entrapped in the horizontal pipes.

Water Supply Connections.—The general arrangement with respect to expansion tanks, and of the water supply connections of heating apparatus, differs somewhat in Great Britain and in America. In the latter country, where a water service is under a suitable pressure, it is the custom to introduce the feed water at a point near the boiler, whilst its height is recorded by an altitude gauge. The expansion tank usually takes a cylindrical shape, and is provided with a gauge glass for observing the water-level at that point. In this country, the cold supply is usually delivered into the expansion tank, an automatic float cock being used for controlling the water supply.

Figs. 10, 11, and 13 show different methods of connecting the overhead feed tanks. The chief advantage of that given in Fig. 10, is that air is readily dislodged from the apparatus when it is charged with water. On the other hand, if overheating occurs, so as to generate steam, the water may be dislodged from the boiler into the expansion tank, when the surfaces of the metal are liable to damage through being burned.

The feed pipe in Fig. 11 is joined to the highest part of the flow, and the advantages derived from this connection are as follows:—

(1) The feed pipe acts as the principal air relief.

(2) If overheating takes place any steam or excess of pressure is immediately relieved without displacing the water from the boiler.

(3) If stop valves are used on the circuits, the feed pipe may be arranged to afford the necessary relief should they be left closed when a fire is lighted.

The principal drawbacks likely to arise are: Some of the pipes may get partially air locked when charging a system, and undue loss of heat from the expansion tank may occur through its contained water being raised to a high temperature. The latter defect may be greatly minimized by protecting the tank with a good insulating material, or the circulating of the water between it and the flow pipes may be avoided by the method

adopted in Fig. 13. In the latter case the feed pipe is trapped, but the merits associated with Fig. 11 are retained.

Fig. 15 shows the feed pipe joined with a drop pipe, instead of with the upper horizontal pipe. In general, it may be associated with the same defect mentioned in connection with Fig. 10, whilst to some extent the same merit is retained. There are other points to which the supply pipe may be attached, but the principal features likely to arise have been already considered. For example, the joining of the feed directly with the top of the boiler is similar in its effect as if it were connected with the flow pipe. In like manner, a direct connection with the lower part of the boiler is similar to joining the supply with a main return.

Expansion Tanks.—When automatic cocks are used for regulating the water supply they should be arranged to close when the water in the tanks is only a few inches deep, the remaining space being utilized for accommodating the expanding water. The levers of the automatic cock should be strong and rigid, that they may easily bear the strain when the floats are submerged. Fig. 16 shows an expansion tank where the water supply is delivered directly into a system from a service pipe, or by the aid of a pump. The tank is joined at the highest point either with a main or with a riser.

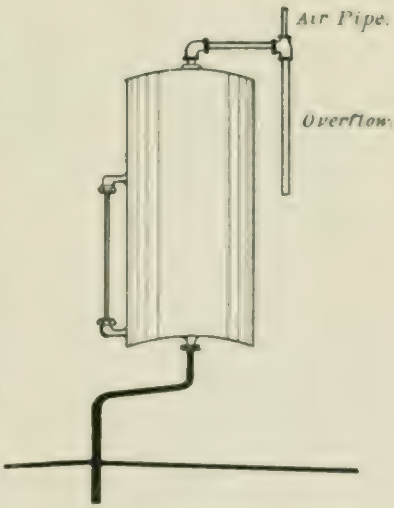


Fig. 16.—Expansion tank.

Fittings and Other Accessory Apparatus.—To aid in the even distribution of fluids throughout a system, special forms of fittings may be used. In some cases, these may be found useful, but generally speaking, if a system is properly designed they offer no decided advantages over the ordinary fittings.

Fig. 17 gives a special form of tee for a main or branch circuit, but for gravity circulating apparatus, a fitting of this class should be used very sparingly, owing to the resistance the projecting lip will introduce. If, on the other hand, the circulation is forced, there is very little objection to its freer use.

A special tee is also designed for risers so as to favour the lower heating surfaces where the circulating head is the least. The chief point in its favour is the neat or compact form it takes. As regards the common fittings, such as elbows and bends, long curved ones are the best to use, whilst the ends of the pipes should be properly reamed after being cut and screwed.

A failing that occurs through the use of unsuitable dimin-

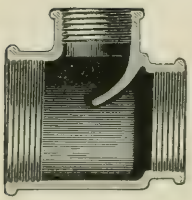


Fig. 17.

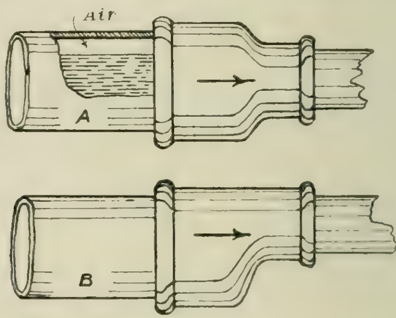


Fig. 18.

ishers is indicated at A, Fig. 18, where air is confined so as to reduce the effective area of a pipe. This defect, however, is easily avoided by pitching the pipes, and by the use of eccentric fittings as indicated at B, Fig. 18.

Air Pipes.—Where convenient, it is a good practice to ventilate the flow risers of a gravity system by air pipes, for by so doing, less air cocks may be used, and a system is the more readily freed from air. As a rule, single air pipes should not be smaller than $\frac{3}{8}$ inch bore, and where an overhead horizontal air line is used to intercept the vertical pipes, care should be observed that it is not rendered useless through getting trapped at some point.

Air Valves.—For hot water apparatus air valves take different forms, but they may be classified under two heads as

hand controlled, and automatic ones. Automatic valves are not free from mechanical defects, but they are the best appli-

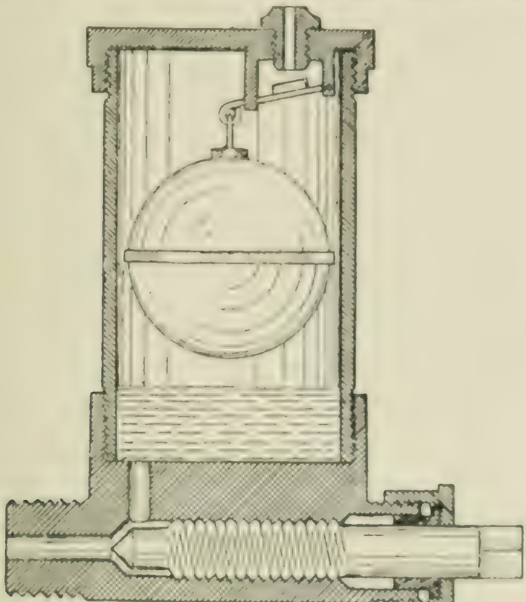


Fig. 19.—Automatic air valve.

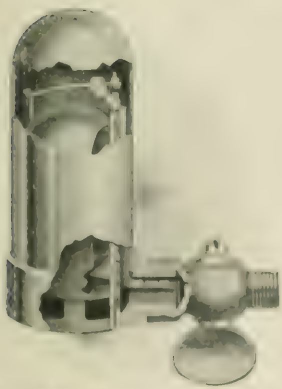


Fig. 20.—"Ideal" automatic valve.

Fig. 21.—"Norwall" automatic valve.

ances as yet devised where air must be discharged from circuits as it tends to accumulate. Fig. 19 gives a common form of

automatic valve, and Figs. 20 and 21 show other styles. For its action, Fig. 19 depends upon the air dislodging the water from the chamber shown, when the ball falls by its own weight and allows the air to escape. The screwed plug at the base of the appliance acts as a stop valve, and can be used as such when it is necessary to effect any repairs. Fig. 20 operates in a similar way, but is an improved type.

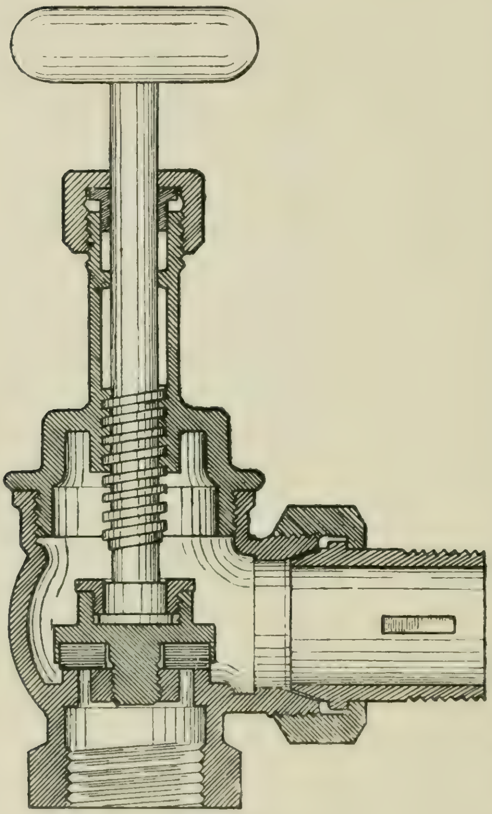


FIG. 22.—Angle valve.

The valve shown in Fig. 21 is intended either for hot water or steam apparatus. When used on the former, the pressure of the accumulating air exerts its force on the top of the float and depresses it, and in turn the orifice is opened for the air to escape. Upon being relieved the air has its place taken by water, when the float is again buoyed up to close the orifice. This valve is more suitable for steam systems.

Stop and Radiator Valves.—When stop valves are used on circuits, fullway or the gate type should be adopted. Globe pattern valves are unsuitable for this purpose owing to the resistance they incur. Radiator valves take different shapes, and if reference is made to the catalogue of a good firm of heating apparatus manufacturers, most patterns will be found to

meet the conditions that arise in general practice. The angle valve, Fig. 22, is a very convenient one, as it permits of a simple and direct connection being made between the radiators and the supply pipes.

A quick-opening valve is shown in Fig. 23, and this being provided with an index plate indicates the extent to which it is opened. It is made in the angle as well as in the straight form.

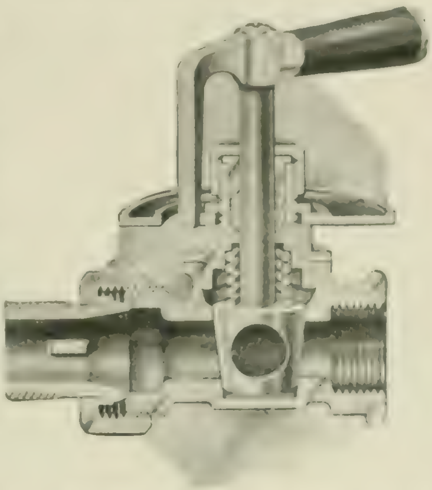


FIG. 23.—Quick-opening regulating valve by National Radiator Company.

When open, both the radiator valves already shown depend upon stuffing boxes for their water-tightness, and as more or less trouble through leakage occurs at these points, valves have been designed with a view to remove this source of weakness. These have been produced in both the slow and quick opening types, the former being indicated by Fig. 24 and the latter by Fig. 25.

The principal feature of Fig. 24 is the prevention of the fluid coming in contact with the valve steam by means of the metal bellows, one end of which is secured to the bonnet A, and the other to the disc holder F. From the construction of the valve it will be seen that the twisting of the stem imparts a vertical motion to the bellows, either to expand or to contract it, according to whether the valve is being closed or opened.

Joints for Copper Pipes.—No great difficulty is involved in the jointing of iron and steel pipes, but the joints of light copper pipes have given trouble from time to time. Owing to the rapid corrosion of iron and steel by certain classes of water, copper pipes are superseding those made of these metals, especially when of small bore and in first-class work. So long as copper pipe of heavy gauge was used no difficulty occurred in

jointing, but thin pipes with screwed joints form a source of weakness. On thin copper pipes only fine threads can be cut, whilst to

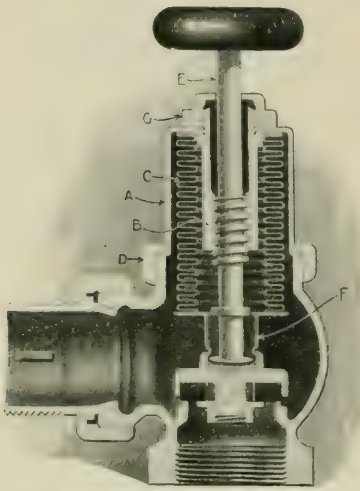


FIG. 24.—" Sylphon " packless valve.

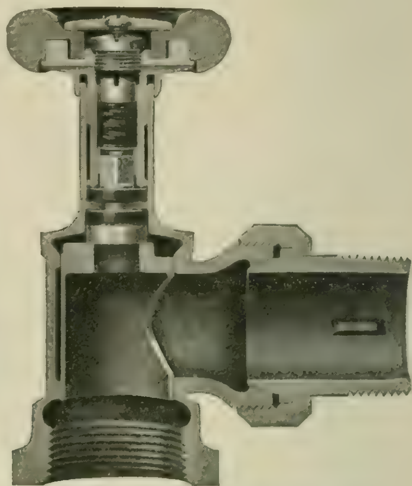


FIG. 25.—" Triton " packless valve. By United States Radiator Corporation.

strengthen the joints it is the usual practice to sweat the pipe ends and fittings together with fine solder. The solder, however, is often responsible for the defect, a galvanic action being

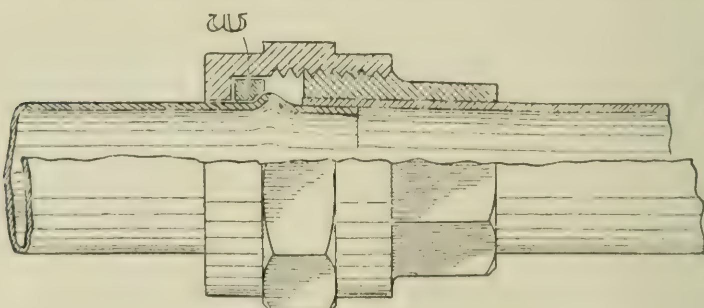


FIG. 26.—Leigh compression joint for light copper pipes.

set up between it and the copper, when the solder is corroded and the joints begin to leak.

At the present time, the best way to fit up thin copper pipes is by the use of compression joints, in which soundness depends upon metallic contact, instead of the use of packing materials. The chief drawback associated with compression joints arises through the special fittings required, and to their higher initial cost. Fig. 26 gives Leigh's joint, in which one pipe end is expanded, whilst the other is slightly tapered, with a bead also formed upon it. For preparing the pipe ends

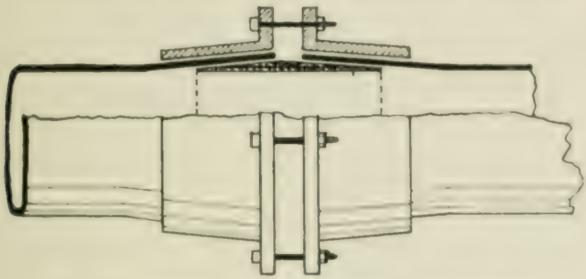


FIG. 27.—Compression joint for light copper pipes.

special tools or machines are necessary, and when this is done they are firmly drawn together by the screwed cap and sleeve piece, the washer W preventing the bead from being damaged.

Another form of compression joint is given in Fig. 27. In this case, both ends of the pipes are expanded and drawn over the tapered ferrule by the flanges and bolts. In comparing these joints, it will be observed that Fig. 27 has an extra point at which leakage may arise, but it is the simpler of the two, and can be used for a greater range of thicknesses than Fig. 26. It is only with pipes of a light gauge that the bead in the Leigh joint can be formed without in some measure reducing the substance of the material.

CHAPTER IV

SMALL-BORE GRAVITY APPARATUS

SMALL-BORE apparatus may take either a so-called "high pressure" or "medium pressure" form. The terms used are relative ones, but it does not necessarily follow that the working pressure in the one will be greater than that in the other. The chief distinguishing feature is this: the "high pressure" form is hermetically sealed, provision being made for the expansion of the water by special tubes, whilst in the "medium pressure" system loaded relief valves are employed.

Systems of Piping.—There are three different ways of arranging the piping of small-bore apparatus: (*a*) on the single circuit

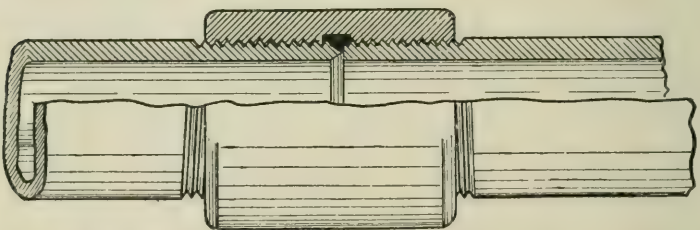


FIG. 28.—Left and right screwed joint for small-bore heating apparatus.

principle, (*b*) on the branched circuit principle, and (*c*) on the crossed circuit principle. The first method is not adopted for large buildings as a rule, for where two or more independent circuits are formed, an expansion tube and charging point would be essential for each. Branched circuits necessitate the use of stopcocks for regulating circulation, and special cocks are often required at certain points for charging the apparatus with water. Crossed circuits are better suited for larger buildings, and although this arrangement nominally divides a plant into a

number of independent units, yet the crossing has the effect of producing one long continuous circuit.

The small-bore apparatus was invented by Mr. Perkins of London in 1831, and is known as the Perkins system of high-pressure heating. The tubes entering into the construction of the apparatus are very strong, lap welded, and approximately of $\frac{11}{16}$-in. bore. Before leaving the works these tubes are subjected to a hydraulic test of 4000 lbs. per square inch, and are put together with left and right screwed joints. The threads

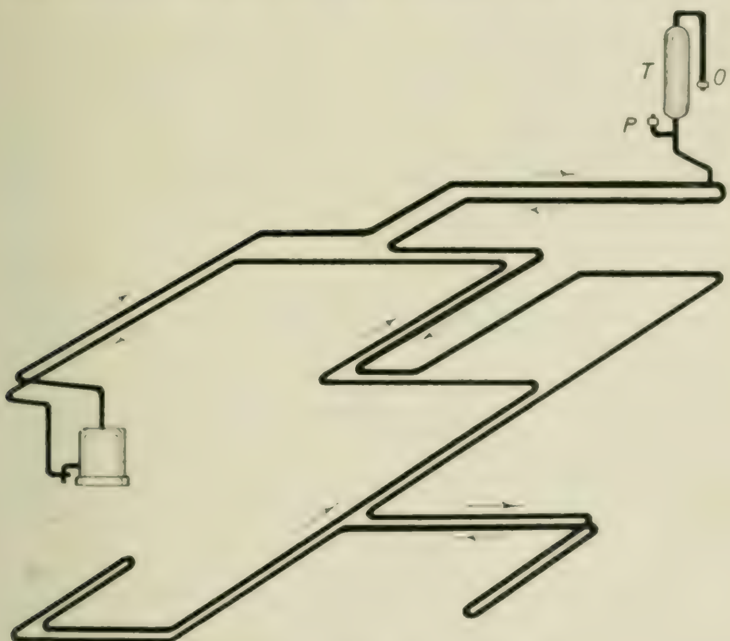


FIG. 29.—Single circuit, high-pressure heating.

are cut with a smaller pitch than that adopted for ordinary wrought iron and steel pipes, and when threading the pipe ends one is prepared with a square flat surface, whilst the other is shaped with a sharp edge. No jointing material is used, the two pipe ends are simply drawn together by a union socket with powerful pipe wrenches until the one cuts into the end of the other, as in Fig. 28.

Although the small-bore system was largely adopted formerly for warming buildings, it has for this purpose been superseded by low-pressure apparatus. It has, however, a large sphere of usefulness in industrial concerns, such as for drying rooms, heating bakers' ovens, heating water, and other purposes.

In Fig. 29 a single circuit is shown. The sealed expansion tube T is located at the highest point, its size being governed

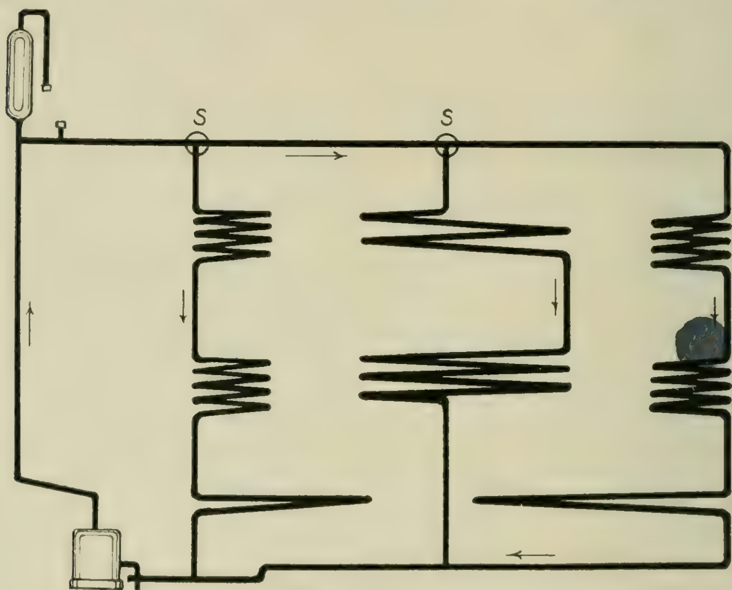


FIG. 30.—Branched circuits, high-pressure heating.

by the capacity of the piping and the temperature to which the water is raised. There is considerable freedom as regards the arrangement of the piping, but the flow portion should be run as directly as possible to the highest point and kept free from dips. If dips are necessary, these should be formed in the return piping.

Under ordinary circumstances the water temperature in a small-bore system is raised to about 300° or 350° F., but when a circuit is of considerable length there is a very pronounced difference in its temperature when leaving and when re-entering the furnace. It is essential to add a little water from time to

time, owing to loss through the porosity of the material, and as too little water causes the circulation to be broken, this state of affairs is soon indicated by the noise created. Water of course can only be added when a system has cooled down, the plugs O and P (Fig. 29) being removed and the water poured in at the latter point.

A branched circuit system is indicated in Fig. 30, and to distribute the water more or less evenly through the piping stopcocks, S, are used. These stopcocks are of special design, and when they are used to control two circuits from one point,

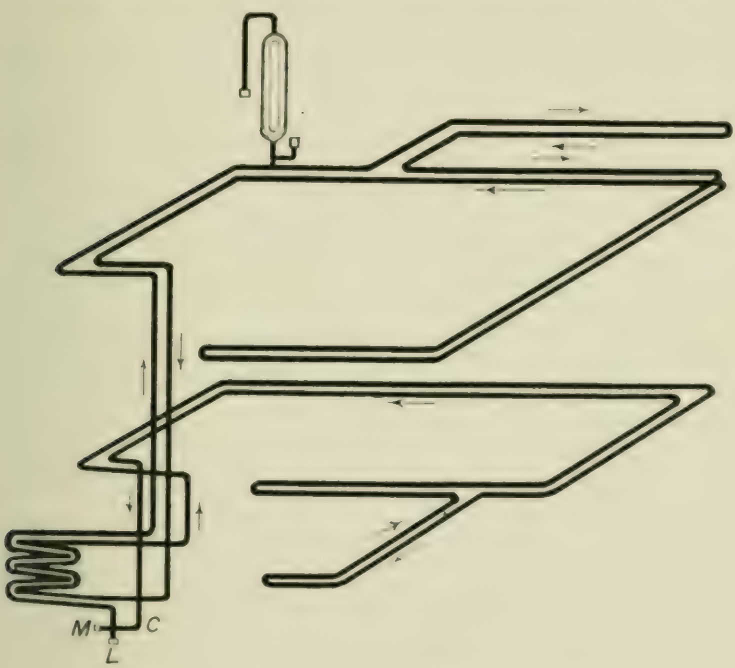


Fig. 31.—Crossed circuits, high-pressure heating.

only one circuit at a time can be put out of use, or the flow of water may be divided between them. It is imperative that a path be always provided through which the water can flow.

Fig. 31 illustrates a system in which the circuits are "crossed." Although only two circuits are shown, the principle

introduced is the same irrespective of the number of units in which a plant may be divided. The furnace coils are bent into any suitable shape, and each acts as the heater for its particular circuit. If the pipes are traced it will be seen that the flow pipe of the upper coil joins with the upper circuit, whilst the return of the upper circuit joins with the lower coil. In a similar manner the flow from the lower coil of furnace joins with the flow of the lower circuit, whilst the return is connected with the upper boiler coil.

Not only are crossed circuits advantageous in distributing the heat more effectually over a building, but the charging of a system is simplified, and only one point for the location of expansion tubes is essential.

When charging Small-bore Apparatus it is necessary to introduce the water by a pump. A special fitting is located, as at C, Fig. 31, and is so arranged that water upon entering at L passes through the whole of the piping before overflowing at M. The fact that the water is made to flow from the pump in one direction is advantageous in that the whole of the air is dislodged from the piping.

Furnaces.—Although it is customary to make the furnace coils of the same tubes as those used for the circuits, still, tubes of a larger bore are sometimes used, but it is essential to avoid all weak points when these are introduced. The furnaces may be of iron or of brick construction, the former as a rule being used for small apparatus, and the latter for larger installations.

The size of a boiler is dependent upon a number of points which will be considered in a later chapter, but so far as the furnaces for small-bore apparatus are concerned, the length of coil to produce a given heating effect is often expressed as a fraction of the total length of piping. It is only a rough-and-ready method, however, but it is a convenient rule when only approximate values are required.

TABLE I.

LENGTH OF FURNACE COILS FOR SMALL-BORE APPARATUS.

Temp. of room Fahr. deg.	Proportion of tube in furnace to total length in circuit.	Temp. of room Fahr. deg.	Proportion of tube in furnace to total length in circuit.
50 to 75	$\frac{1}{10}$	95 to 140	[illegible]
75 to 95	$\frac{1}{8}$	140 to 200	[illegible]

Size of Expansion Tubes.—Ample provision should be made for the expansion of the water so as to obviate damage through overstrain. Water does not expand at a uniform rate, but is greater in high than in low temperatures. The expansion, however, for very high temperatures does not appear to have been accurately determined, but the following values may be used in the design of heating apparatus.

TABLE II.

APPROXIMATE EXPANSION OF WATER BETWEEN 40° AND 600° FAHR.

Temp. deg. Fahr.	Volume.	Temp. deg. Fahr.	Volume.
40	1·0000	400	1·1484
212	1·0433	450	1·1843
300	1·0869	500	1·2283
350	1·1156	600	1·3000

Expansion tubes often have a bore of 3 inches, but other sizes may be used, and where the length of one would be unwieldy, two or three shorter tubes may be joined to give the requisite capacity.

For determining the length of expansion tube the following formula may be used—

$$L = \frac{cl}{d^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where L = length of expansion tube in feet,

l = total length of tube ($\frac{3}{4}$-inch bore) in circuits, heating coils, and furnace in feet,

d = internal diameter of expansion tube in inches.

c = a coefficient which varies with the maximum water temperature.

For a maximum temperature of			300° F.	$e = 0{\cdot}08$
,,	,,	,,	350° F.	$e = 0{\cdot}10$
,,	,,	,,	400° F.	$e = 0{\cdot}14$
,,	,,	,,	450° F.	$e = 0{\cdot}18$
,,	,,	,,	500° F.	$e = 0{\cdot}22$
,,	,,	,,	600° F.	$e = 0{\cdot}30$

Example 6.—Assume an installation consisting of 540 feet of small-bore tube for a maximum water temperature of 350°, what length of 3-inch diameter expansion tube would be required?

By formula 3—

$$L = \frac{el}{d^2}$$

Substituting values $\quad L = \dfrac{0{\cdot}1 \times 540}{3^2}$

when $\quad L = \textbf{6 feet.}$

Medium Pressure Small-bore Apparatus.—There is no difference in the design of this and that of the "high pressure"

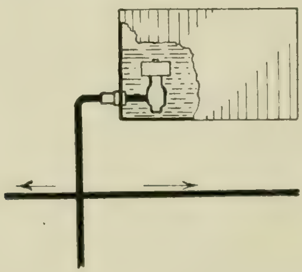


FIG. 32.—Expansion tank for small-bore apparatus.

arrangement. The only departure is one of detail, an open tank and loaded valve taking the place of the expansion tube. Fig. 32 shows the tank and valve for joining at the head of a circuit, whilst a section of the valve is given in Fig. 33. It takes a combination form, the upper part being loaded according to the pressure to be carried, whilst a vacuum valve is formed at the lower part. This device permits of the expanding

water escaping into the tank, and returning to the apparatus as a cooling action sets in.

The small-bore heating system is quick in action owing to its small water capacity, and it is comparatively cheap to install.

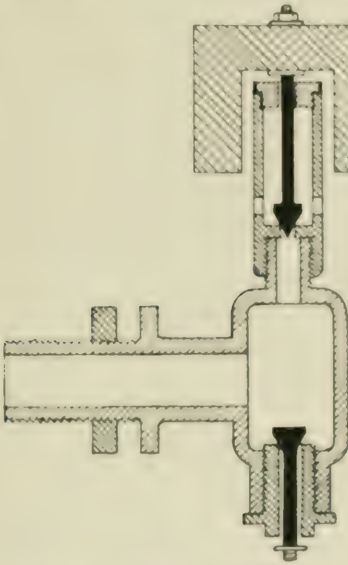


FIG. 33.—Relief and vacuum valve.

When an apparatus is composed of $\frac{13}{16}$-inch bore tubes its approximate capacity in imperial gallons may be obtained by dividing the total length of piping by 44. If its capacity in American gallons is desired, the total length of piping should be divided by 36·6, or say 37.

CHAPTER V

ACCELERATED HOT-WATER CIRCULATING SYSTEMS

WHERE accelerated circulation is simply due to a high water temperature, the latter is usually obtained by some contrivance that partially seals a system. Under such circumstances an installation resembles the medium pressure small-bore apparatus, excepting that the ordinary form of piping is adopted instead of tubes of small diameter.

Sealing devices can generally be applied to any ordinary system; they may be useful for increasing the heating capacity of an existing installation, and they are comparatively cheap. To be a success, however, ample boiler power is essential.

The most popular means of partially "sealing" a heating system at the present time is by mercury, the appliance used taking a simple form, but an apparatus cannot be subjected to a greater pressure than that for which the device is designed. Another feature of a mercury seal is that the apparatus on which it is employed comes within the category of a low pressure one. On the other hand, small-bore apparatus and those in which loaded valves are used, in order to conform with the provisions of the London Building Act, require the piping fixed three inches clear of woodwork and other inflammable material, whilst no such restrictions apply to low-pressure systems as generally defined.

When an apparatus is open to the external air, the boiling point of the water at the highest level is dependent upon atmospheric pressure, the latter varying with altitude and the prevailing meteorological conditions, whilst the normal boiling temperature at sea level is 212° F. In a heating system the boiling temperature varies at different elevations, increasing as the hydrostatic pressure increases; if, therefore, in a boiler,

water is raised to over 212° F., the excess heat beyond that coinciding with atmospheric pressures will, unless absorbed by the lower heating surfaces, cause the production of steam when the water reaches the highest level. By partially sealing a system, however, the boiling point is raised, and as the resistance due to the sealing device is usually equivalent to a pressure of 10 lb. per sq. inch, the water, even at the highest part, may have its temperature increased to within a short distance of 240° F. The latter value is the boiling point for the pressure given. Thus, the water between leaving and re-entering a boiler may be subjected to a considerable temperature drop, which affects the rate of circulation. Where it is desired that the fall of temperature shall be further increased, this can be done by increasing the resistance of the sealing appliance. Figs. 34 and 35 give two mercurial sealing devices for accelerating circulation, and although they differ somewhat in form they operate precisely in the same way. Into the lower chamber of each appliance mercury is poured until it reaches the level of the small plug on the right, thus sealing the lower end of the double tube which communicates between the lower and upper compartments. The supply or expansion is joined with the top connection, and the bottom one is attached to some other part of the system, the precise point of junction depending upon the piping adopted. Assuming the water in a system is cold and the surface of the mercury at its normal level, upon the application of heat the expanding water presses on the mercury, dislodging it through the double tube to the upper compartment of the fitting; at this stage the increased volume of water passes to the expansion tank, whilst the mercury endeavours to return to the lower level; in fact, a partial circulation of the mercury is set up within the double tube through permitting

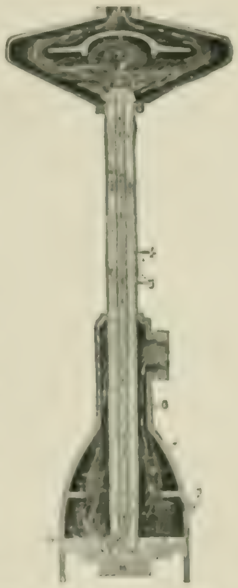


FIG. 34. "Honeywell" heat generator.

the excess water to flow to the expansion tank. When contraction occurs, very little opposition is offered to the return of the water, and what resistance there is, is represented by the depth of the mercury at the bottom of the appliance. To overcome this, the expansion tank only requires locating a few feet higher than the sealing device, the minimum distance being about 3 feet.

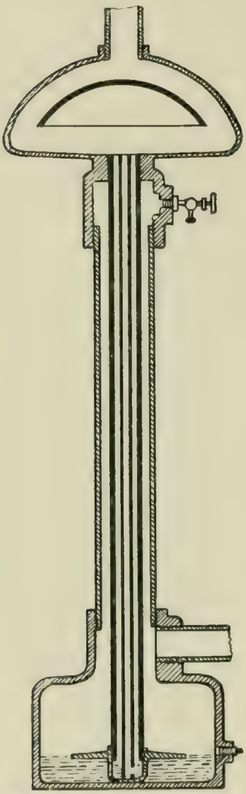


FIG. 35. — "Klymax" heat generator. By Kellogg, Mackay, & Co.

When comparing Figs. 34 and 35 it will be observed that the principal difference between them is simply one of detail, special provision being made in Fig. 35 for dealing with the air. In Fig. 34 it would be necessary for any air to be forced through the mercury seal, unless a special air chamber were used. Air pipes of course cannot be adopted on circuits when sealing appliances are used, although special automatic valves may be fixed for effecting the escape of air.

For an "overhead" or "drop" system the usual point of connection with a mercury seal is shown in Fig. 36, whilst that for an up-feed system is indicated in Fig. 37. The latter position is the better of the two, for the appliance works more smoothly, and there is less likelihood of the mercury being precipitated against the curved deflection plate.

In some cases, combination relief and vacuum valves take the place of mercury seals, but where these are adopted, a type should be selected that is sensitive and reliable in action. For ordinary heating systems, the combination type shown in Fig. 33 is not suitable, a superior arrangement being one where the vacuum valve opens by its own weight when the water tends to leave it. The chief advantages of valves are due to the facilities they offer in the way of adjustment with respect to a larger range of working pressures.

Some of the literature on these appliances greatly overrate their value, but the extent to which the circulation is accelerated may be computed by formula 2.

A recent application of the mercury seal is shown in Fig. 38, and the important features of this system are, the location

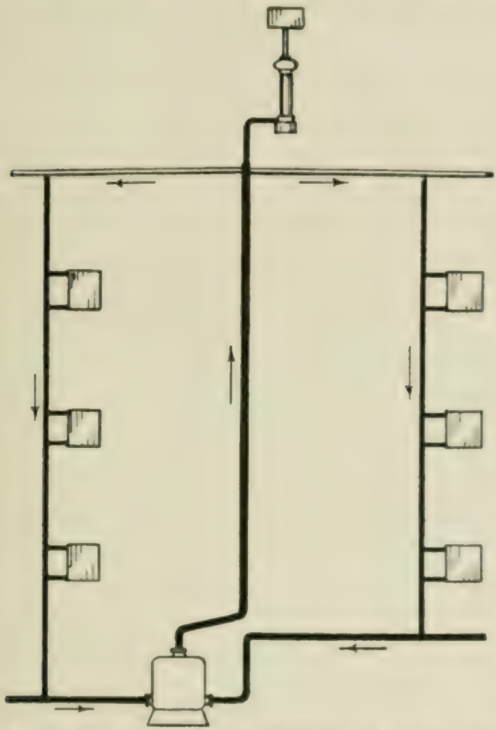


FIG. 36.—Showing position of generator for "drop" system.

of the expansion tank, the relief of any excess pressure, and the means adopted for regulating the rate of combustion. The mercury trap is joined with the return on the left, the seal being just sufficient to hold back the pressure due to the head of water plus an additional 10 lb. per sq. inch. To the top of the flow-pipe bend, the expansion tank is joined with a pipe of small bore, and from one end of the tank, another pipe communicates with the diaphragm that operates the check and draught dampers of the boiler.

When a system like Fig. 38 is charged with water, the air in the tank is compressed, and this is subjected to still greater pressure as the temperature of the water is raised. Whatever pressure is created in the tank is at once transmitted to the diaphragm, whilst, by means of the perforated plate at the end of the lever, the dampers may be set to operate at any

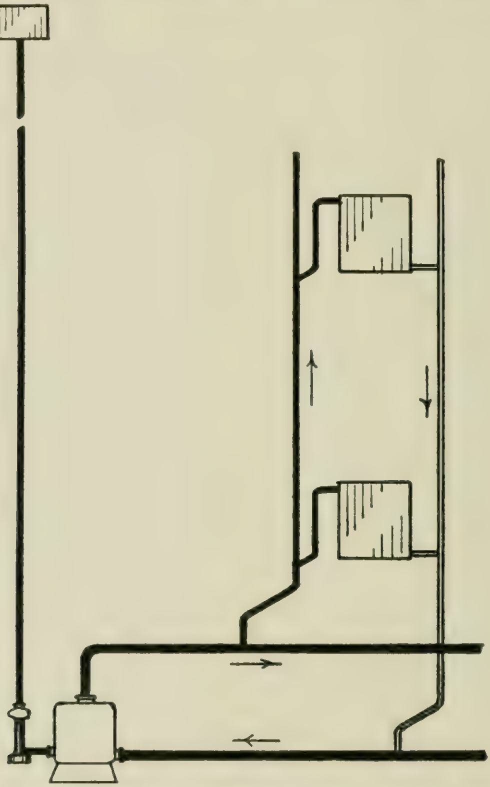


FIG. 37.—Showing position of generator for "up-feed" systems.

temperature desired. Should the diaphragm fail to act at any time, any excess of pressure is immediately relieved by water escaping through the mercury seal. The water may be added by joining a service pipe directly with a return, and where this form of connection is not permitted, a hand pump could be used.

Another system in which the expansion tank is located in the boiler house is given in Fig. 39, and this presents a novel feature as regards the boiler draught control. Here it is preferable to place the expansion tank alongside the boiler, as

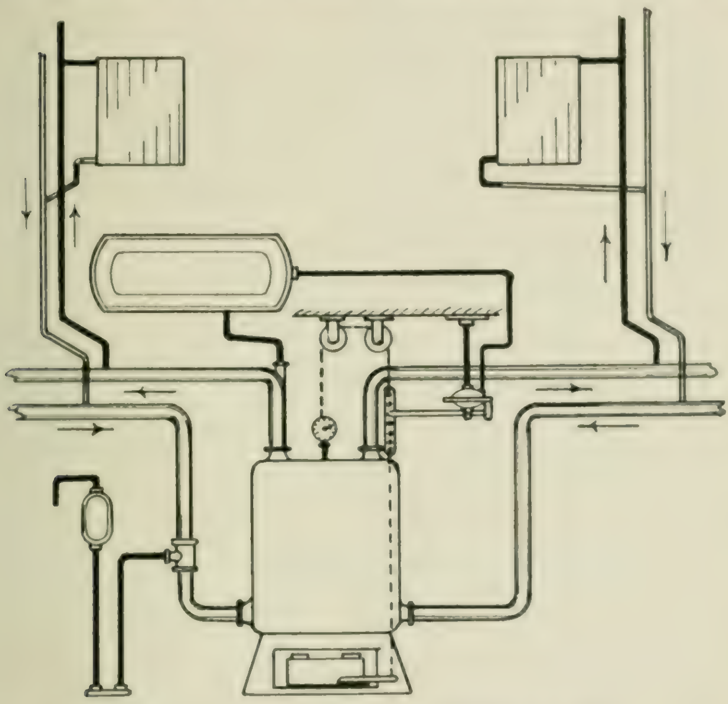


FIG. 38.—Accelerated system by Dongherly & Tabler.

the former is arranged to fall bodily and to rise through a short distance to impart motion to the lever. To the expansion tank is attached a balance weight, and for normal working conditions the contained air and water are proportioned to keep the tank in the higher position. If, however, the temperature continues to rise, further water enters the expansion tank, when, by virtue of the additional weight, it falls to the lower position by the aid of the gland joint shown. Upon a cooling action being set up, the water is displaced from the tank by the compressed air, when the tank is again raised by the balance weight, and the position

of the dampers is reversed. The gland joint only permits of a half-inch drop, but this distance is multiplied for operating the dampers by means of the lever arm. At the top of the expansion tank, a relief valve is provided.

The mode of operation and method of charging Fig. 39 is as

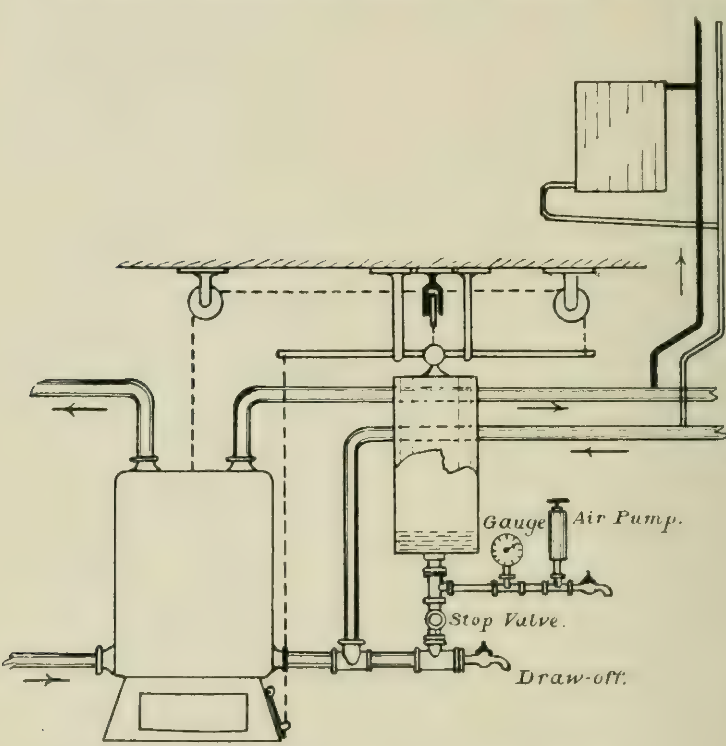


FIG. 39.—Accelerated system of hot-water heating.

follows: Before filling the apparatus with water, the expansion tank is disconnected by closing the stop valve shown, after which the whole of the pipes and radiators are charged in the usual way. Air is then pumped into the tank until sufficient pressure is produced to hold up the water in the highest heating surface, after which a few gallons of water are withdrawn to allow for the expansion that accompanies the application of heat. This being done, the stop valve is opened, when the

water should rise to a predetermined point. It is more convenient if the gauge used only records the pressure that accrues

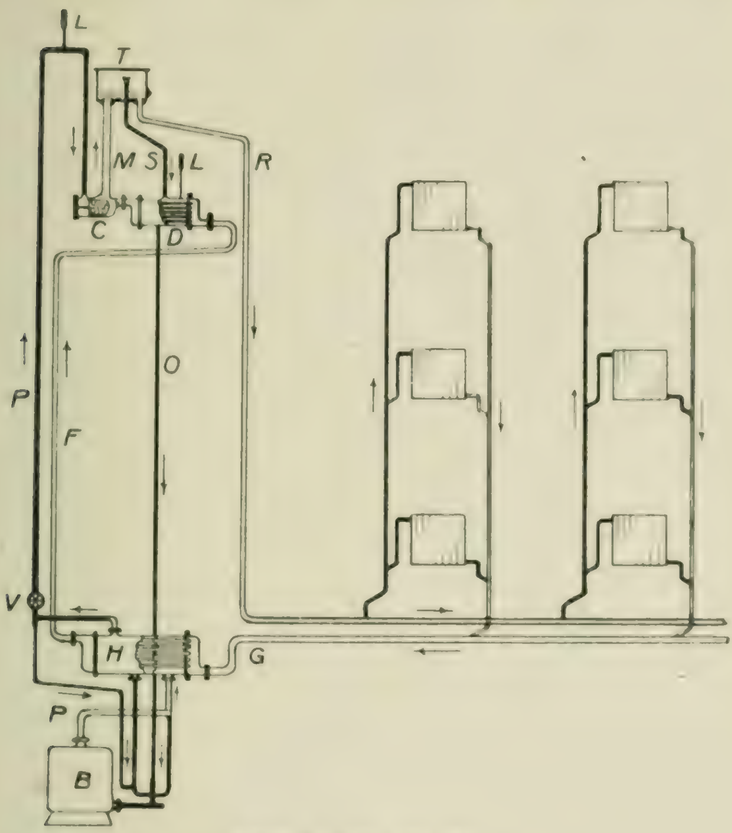


FIG. 40.—"Reck" system

L = automatic vent.
T = expansion tank.
S = overflow and exhaust pipe.
R = return from tank.
M = mixing pipe.
C = circulator.
D = condenser.
O = condensation return.
P = steam pipe.
V = regulating valves.
F = flow pipe.
H = reheater.
G = return from heating surfaces.
B = low-pressure steam boiler.

from the heating up of the system, it being graduated for the index finger to point to zero when the water is cold, irrespective of what the static pressure may be. By the adjustment of the

balance weight, the dampers may be set to operate at any pressure desired.

In Fig. 8, page 28, the general principle of acceleration by the aid of steam is shown, whilst Fig 40 gives an apparatus on the "two pipe" principle in greater detail. An ordinary low-pressure steam boiler is used, which should be fitted with an automatic draught regulator, but the reheater H as a rule is discarded. For the apparatus to be a success the steam that escapes into the expansion tank must be readily condensed, or there will be a tendency to the equalization of the pressures in the pipes RG and MF, when the circulation will be greatly impeded.

Although in Fig. 40 a surface condenser is used to ensure the necessary differential pressure, the surplus steam may also be condensed by being brought into direct contact with the cooled return water, especially in those cases where no reheater is used. To effect this, a condensing tank is placed at a certain point between the expansion tank and the circulator, the top of the former being joined with the overflow from the expansion tank, whilst the pipe from the bottom of the condensing tank is taken to the circulator. The return pipe from the heating system is also connected with the condensing tank so as to bring the steam and water in direct contact. The expansion tank should be properly proportioned and charged with water to a certain point, any deficiency through leakage or other cause being indicated by the gauge at the boiler. If desired, the risers may be vented to the atmosphere.

Where acceleration is due to the introduction or disengagement of steam, it is sometimes considered a drawback, in that the water is delivered in a highly heated state to the radiators. This point, however, is easily overrated, for the water, upon entering the heating surfaces, at once mixes with the cooler water and so lowers the temperature.

Fig. 41 gives a form of rapid circulating plant where acceleration is due to the liberation of steam bubbles in the ascending water columns. In an installation of this type it is necessary to limit the production of steam, otherwise it would be noisy and somewhat erratic in operation. The principal

features in Fig. 41 are the provision of the "flow bottle," the method of operating the draught regulator, and the large "dip" or siphon that forms part of the expansion or relief pipe. By opening the valve V, the system works in the ordinary way, the increment due to expansion having direct relief whilst any air that is liberated passes directly to the expansion tank.

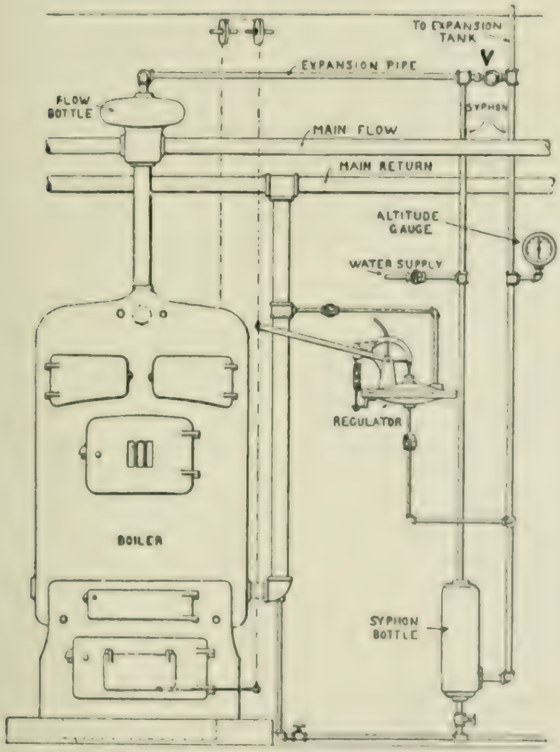


FIG. 41.—"Beck" system.

When the valve V is closed, the direct connection is cut off, and the only course left for relief is through the siphon or dip. As steam rises from the boiler it tends to gather in the "flow bottle" instead of being conducted into the horizontal pipes, and in this way its contained water is dislodged through the siphon, thus subjecting the two sides of the regulator diaphragm

to differential pressure. The dampers are automatically adjusted to regulate the generation of steam, but the order of the dampers is again reversed as soon as the steam in the "flow bottle" is condensed. It will be observed that the underside of the regulator is subjected to a constant pressure in virtue of its being joined with the pipe that leads directly to the expansion tank, whilst the pressure on the top of the diaphragm is the variable one.

It is usual with the system indicated in Fig. 41 to locate the "flow bottle" and the horizontal main pipes as high as practicable, and although the steam from the boiler is largely intercepted by the "flow bottle," yet the water in the horizontal flow mains may be maintained at a sufficiently high temperature, that upon its passage into the vertical risers steam will be disengaged. The production of the steam in this manner is due to the diminishing hydrostatic pressure to which the heated particles are subjected in ascending to the higher level, and as a mixture of steam and water gives a diminished density for the ascending columns, the circulation is accelerated throughout the installation.

Still another type of apparatus is indicated in Fig. 42, where acceleration is due to displacement by steam. At the head of the system, tanks T and M are shown, and from each the water is alternately dislodged, and delivered through the circuit, being finally received in the adjacent tank. In the tank T a float chamber is placed, this being arranged as it rises and falls to open one of the tanks to the steam supply, and the other to the exhaust pipe E. The non-return valves A, B, D, and H control the course the water must take, whilst at the same time they admit of the apparatus being operated by natural circulation when this is desired. A system may take different forms, and steam may either be supplied from a separate source, or the hot water boiler may be replaced by a calorifier, and a low-pressure steam boiler installed. The action of the tanks resembles a series of pulsations, but the circuits will require to be properly sized and adjusted for the water to be evenly distributed over the whole plant.

For dealing with the exhaust steam in Fig. 42, a small surface condenser C is shown, and into this the surplus water

is conveyed that accrues from the condensation of the steam in the circulating tanks.

In Fig. 43 the circulating tanks of the last apparatus are more clearly shown, it being assumed that the tank T is open

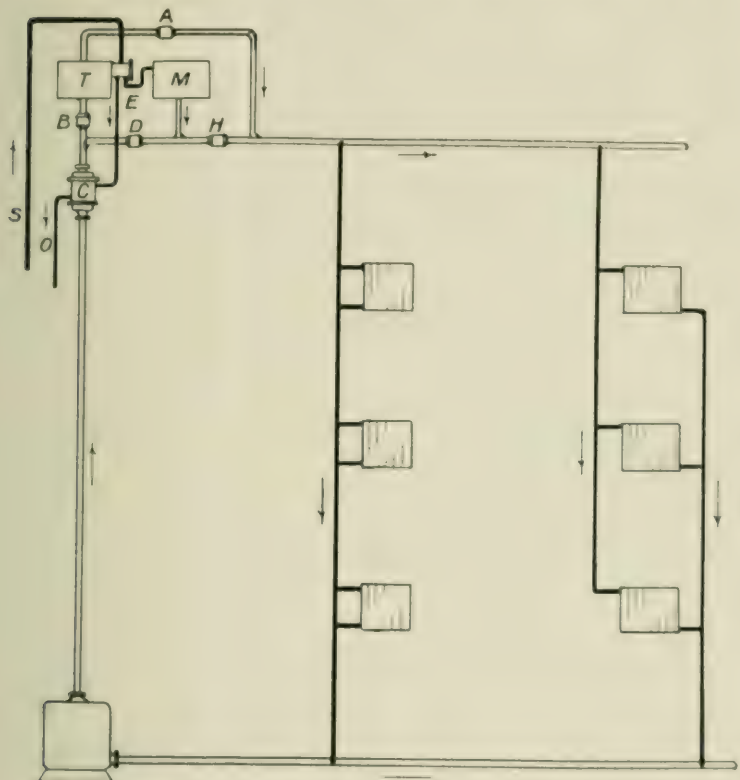


FIG. 42.—Baker's system of circulation.

S	= steam supply.	E	= exhaust pipe.
O	= condensation.	C	= condenser.
T, M	= circulating tanks.	A, B, D, H	= check valves.

to the exhaust pipe, whilst steam under a suitable pressure is being admitted to M. From the latter tank the water would be displaced, and, provided that the valves D and A are in order, would flow through H, complete the circuit and finally enter tank T. When, however, the water in T has filled the float

bucket and weighed it down, the position of the valve is reversed, the tank T being opened to the steam supply and M to the exhaust. Under the altered conditions, the water is now dislodged from the float, completes the circuit as before,

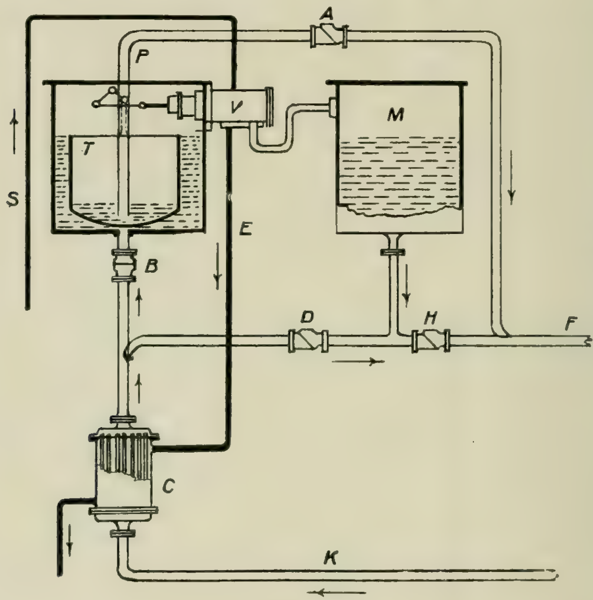


Fig. 43.

and enters M, when the float is buoyed up, and the position of valve again reversed. So far as the speed of the circulation is concerned in this case, it will be chiefly governed by the pressure of the steam and the resistance offered by the piping.

CHAPTER VI

FORCED HOT-WATER CIRCULATING APPARATUS

ALTHOUGH the application of external power to produce a positive movement of water is common to all forced systems, yet they differ in form, such as in the construction and arrange-

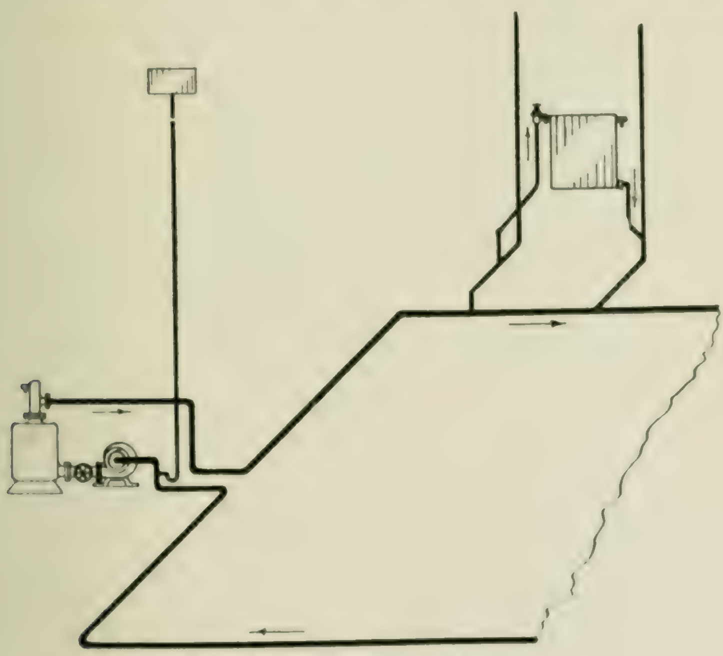


FIG. 44.—Forced circulating system.

ment of the heaters, the manner in which the connections are made, and in the arrangement of the piping. Fig. 44 gives a simple system in which the water is forced through the main

circuit, whilst the movement through the risers depends upon natural or gravity circulation.

Where a simple circuit system is adopted, the movement through the risers may be accelerated by the use of the fittings shown in Fig. 45. Here the nozzle for the flow riser is turned to face the stream, whilst that for the return riser is fixed the opposite way. This arrangement has the effect of converting the velocity into static pressure at the flow, and of producing a partial vacuum at the end of the return riser.

The fittings shown, however, have a restricted use in these

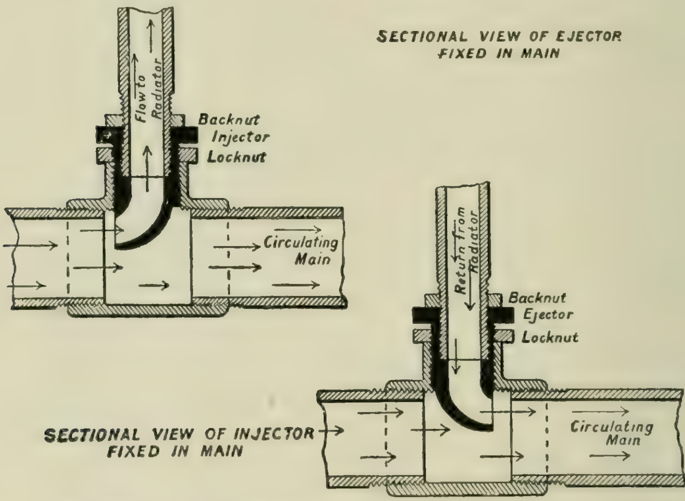


Fig. 45.—" Acme " fittings by National Radiator Company.

systems, and it is more usual in a one-pipe arrangement to depend upon the use of long sweep fittings. In some cases, instead of joining the risers directly to the main circuits, subsidiary loops are formed, these being sized and arranged so that a positive movement of water through them is ensured. To the subsidiary loops the risers are joined. For intercepting the air in Fig. 44 an air vessel is shown, and from this it may be released by a hand-controlled valve, or an automatic air trap may be substituted for the vessel shown. The feed and expansion tank is joined to the return, close to the inlet

side of the pump, but where there is circulating pressure the open tank requires to be replaced by a closed one with automatic feed as in Fig. 7. As a rule, an open tank can be used for all plants where the feed pipes can be joined as in Fig. 44.

Fig. 46 indicates how radiators are "shunted off" the main circuit. If a given circulating pressure is generated by the pump, it is clear that this will be absorbed by pipe friction as

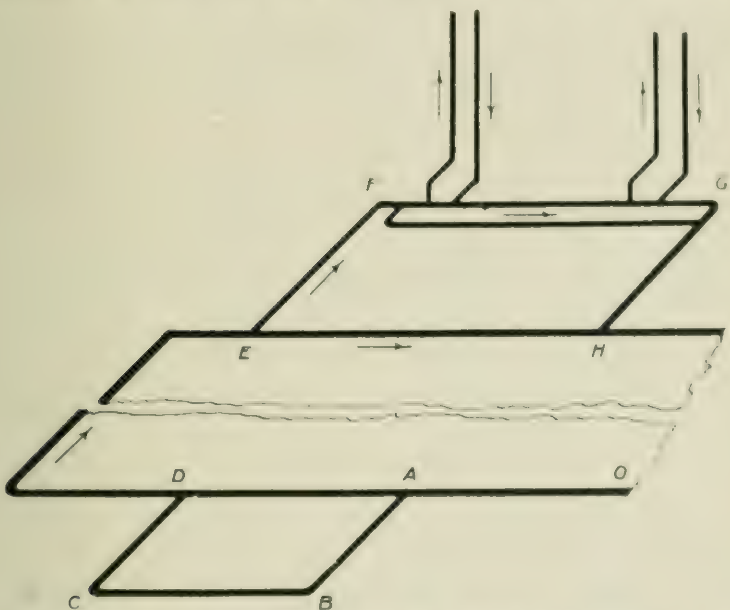


FIG. 46. Forced circulation. "One-pipe" system with loop circuit

the water completes the circuit. Let it be assumed that the circulating pressure at A is equal to a head of 30 feet, whilst that at D is 29·5 feet. In this case the fall of pressure is half a foot, and represents the head absorbed by the length of main between A and D and by the loop ABCD. Whatever weight of water is circulated, the velocity through OA will be greater than that through AD owing to the additional path provided. The proportion of the water circulating through the loop ABCD will depend upon the resistance introduced, but the velocity through

it will necessarily be less than where the water can take the direct course as from A to D. In like manner, the pressure head at E may be assumed as 22 feet, and that at H as 21 feet, thus giving a pressure drop of 1 foot of head. In order to diminish the frictional resistance from F to G, a double pipe is shown, whilst the risers are joined to this portion of the loop.

A portion of a "two-pipe" system with forced circulation is shown in Fig. 47. With this arrangement, a much greater force is available to circulate the water through the branches than in the case of a "one-pipe" or circuit system. The extent of the differential pressure to produce motion in a branch circuit varies with its distance from the pump, and with the initial circulating

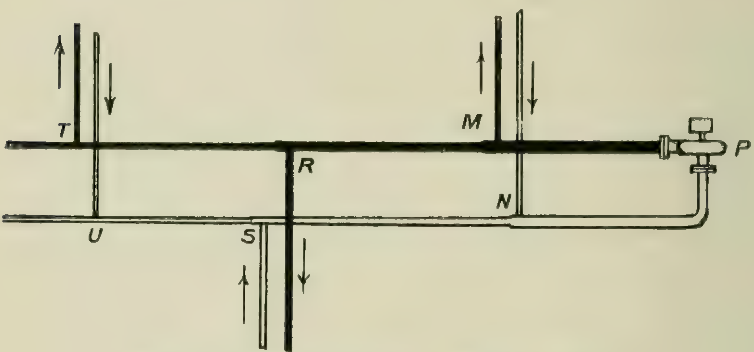


FIG. 47.—Forced circulation. "Two-pipe" system.

pressure. For example, if the water upon leaving the pump has a circulating head of 28 feet, 24 feet at M, 20 feet at R, 16 feet at T, 12 feet at U, 8 feet at S, 4 feet at N, and zero at the inlet of the pump, the pressure difference between M and N would be 20 feet, between R and S 12 feet, and between T and U 4 feet. Thus it will be seen that the velocity of flow will vary considerably in the different branches, and that a given size near to the pump will serve a larger area of heating surface than one further removed, provided the remaining factors are equal.

To regulate the flow of water through the branches of a two-pipe system, some form of throttling device will be essential, for with standard sizes of tubes alone, the resistance of the

various sections cannot be exactly proportioned. For this purpose, valves are sometimes used, or where cheaper means are required, special tees with throttling plugs can be utilized, or orifices of different diameters can be introduced.

For circulating water in a heating system, a centrifugal or turbine pump should be used. Piston types are not so suitable for regulating the speed of circulation, whilst some forms are rather noisy in action.

The more interesting aspect of circulating plants is that in which either "exhaust" or "live" steam is available for heating the water, and for works and other large buildings these systems often open up possibilities for economy and flexibility of operation that are unrivalled in any other form of heating apparatus.

Whether a "live" or "exhaust" steam heater can be the more advantageously employed, depends upon the case as a whole, such as the amount of steam required for warming purposes, the extent to which this varies throughout the heating season, and the type and size of the engines in use. Conditions frequently arise, however, where it is desirable to provide both forms of heaters, the live steam heater being the one used during the milder weather, and when the engines are stopped.

Connections of Heaters.—There are various ways of arranging the heaters and their connections. In Fig. 48, an arrangement is given that is sometimes suitable where exhaust steam is available, either from a condensing or non-condensing engine. Assuming in the first place that a non-condensing engine is in use, the back pressure valve B diverts the "exhaust" to the heater, whilst the separator G will aid in keeping the heater tubes in a cleanly state. To regulate the temperature of the water as it leaves the heater, the back pressure may be varied, or the stop valve adjoining the separator may be used, or the valve on the condensation pipe may be partially opened or closed. The latter procedure results in a portion of the heater tubes being thrown out of use, according to the depth to which the condensation rises in the casing. In order that the water temperature can be maintained when the engines are stopped, or when the "exhaust" is insufficient, provision is made for the admission of "live" steam. To prevent the latter from

escaping into the main "exhaust" when the stop valve is open, a non-return valve V may be added, or a combined stop and non-return may be used.

The introduction of "live" steam as in Fig. 48 is not the most economical method, owing to the whole of the water of condensation requiring to be pumped back to the boilers, and to the loss of heat that accrues through reducing the steam from a high to atmospheric pressure. This connection, how-

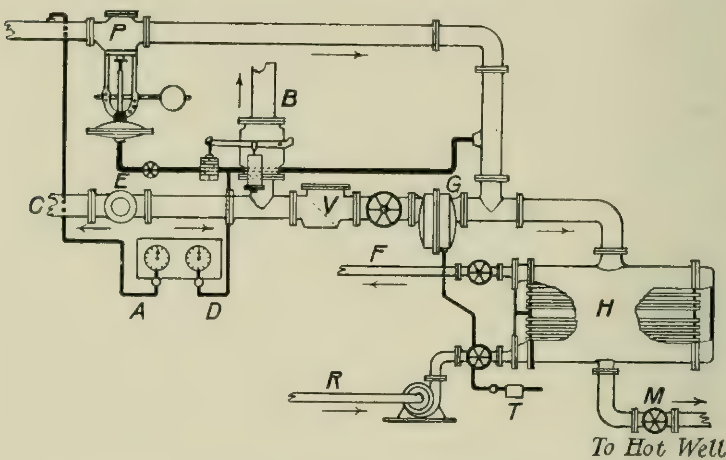


FIG. 48.—Forced circulating system.

P = pressure reducing valve.
B = back-pressure valve.
C = pipe to atmosphere.
E = exhaust from engine.
V = check valve.
G = grease extractor.
F = flow pipe.
R = return pipe.
H = steam heater.
A, D = steam gauges.
T = steam trap.
M = valve.

ever, is suitable where the heater is some distance removed from the source of supply, or where a "live" steam heater cannot be sufficiently elevated to permit of the condensation returning by gravity to the boiler.

Should the "exhaust" steam for a case like Fig. 48 be received from a condensing engine, the economy of the system would chiefly depend upon the relationship between the power and heating loads. This aspect of the problem is considered in the chapter on "Exhaust Steam Heating."

With a condensing engine, the heater and connections as

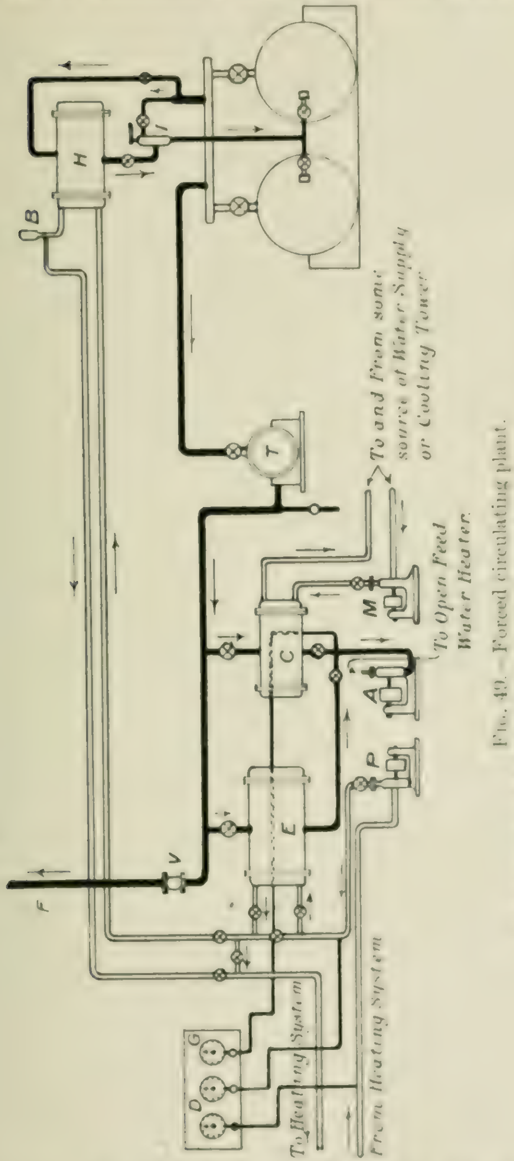


Fig. 49. — Forced circulating plant.

B = automatic air valve.
H = live-steam heater.
I = injector.
F = exhaust.
V = back-pressure valve.
E = exhaust heater.
C = condenser.
P = circulating pump for heating system.
A = air pump.
M = circulating pump for condenser.
D = differential pressure gauges for P.
G = vacuum gauge of condenser.

shown in Fig. 48 may be considered in the light of a supple-

mentary condenser to be used in the winter time, whilst the ordinary condenser is utilized in warm weather and when only small heating loads are required. There would be this difference, however, that whereas the general condenser would be operated under a vacuum, the heater or supplementary condenser would be under a small pressure, which would considerably increase the steam consumption of the engines. When, however, heating and power plants are considered as a whole instead of as separate units, there would be no loss in the combined efficiency, so long as the circulating water of the heating system condenses a certain percentage of the steam.

Fig. 49 gives a combination for utilizing the exhaust from one or more turbines where the average steam required for heating purposes is approximately equal to, or in excess of, that required by the power units. The condenser C and exhaust heater E are arranged so that either one or the other can be put in or out of use, or they may be operated together when the heating load falls appreciably below that of the turbines. In other words, the condenser C would deal with the "exhaust" that the heater E could not condense.

The arrangement of Fig. 49 is advantageous, as the turbines may be operated under a high vacuum for a large variation in the weather conditions. For example, where the "exhaust" is insufficient to supply the heating demand, the water may be circulated through the "live" steam heater H after first passing through the exhaust heater E. It is only a matter of manipulating the stop valves shown, for the temperature of the water leaving the heater H may be controlled by allowing the condensation to accumulate, and to cover part of the heater tubes. On the other hand, where the volume of steam required for heating purposes is rather more than the turbines will supply when operating under the usual vacuum, the steam consumption may be raised to the right amount, either by altering the speed of the air pump, or by admitting air to lower the vacuum. The live steam heater H should have a capacity sufficient to do the whole of the heating work when the turbines are stopped, and provided it is located well above the boiler, the condensation can be returned without the intervention of a pump. It may, however, be essential to provide an injector at I to facilitate the return of

the condensation, when the condenser is working nearly at its full capacity, or where the head space above the boilers is limited. The air pump A is assumed to be of the rotary form, coupled directly with an electric motor, whilst the circulating pumps P and M are of the centrifugal or turbine class, and driven in the same way.

Plants for High Buildings.—When forced circulating systems are used for very high buildings, it is desirable to divide the water circuits into two or more isolated units, in order that excessive strain be avoided at the lower levels. The steam, however, for heating the water may be supplied from a common source, and if exhaust steam is available it may be utilized as far as it will go. This idea is indicated in Fig. 50, where two principal water circuits are provided, each supplying a number of different floors; but additional main circuits may be added in the same way. The engine M is assumed to be operating as a non-condensing one, and the water of condensation from C returned through a feed heater or economiser to the boiler. The arrangement of the valves permits of the separate use of either the "live" or "exhaust" heater, or the water may be circulated first through the one and then the other. It will be observed that the valves on the condensation pipe from the "live" steam heater are located in the boiler house, so that the temperature of the circulating water can be largely regulated from that point.

In forced circulating systems the total pressure at any point is represented by the static plus the circulating pressure, so that if it is desired that the pumps shall not be subjected to the maximum strain they should be located at a fairly high level instead of at the lowest point.

In Fig. 50 the expansion tanks T are located near the pumps, and this position is convenient in that it helps to concentrate the important parts. For forcing the water to the top of a system compressed air may be introduced into the expansion tank from any convenient source, or the pressure may be generated by the aid of a force pump. Each expansion tank should be provided with a gauge glass to indicate the volume of water it contains, whilst a pressure gauge and relief valve should also be added. As regards the piping for supplying the

radiators, this may be arranged on either of the recognized

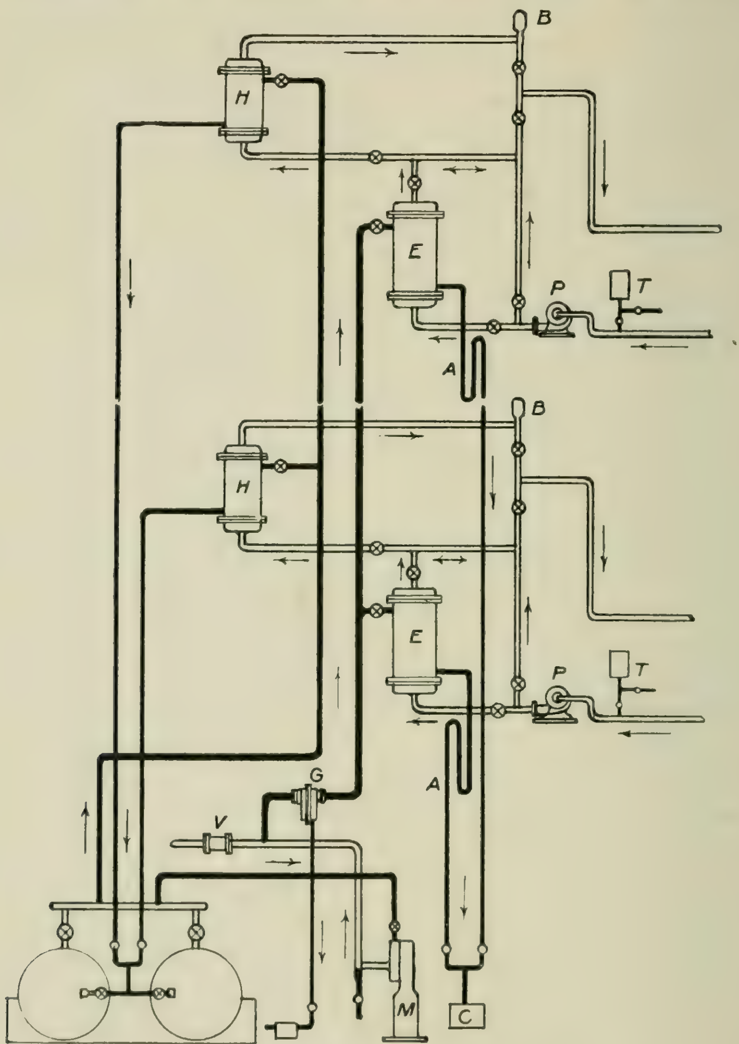


FIG. 50.—Forced circulating plant for high buildings.

B = automatic vent.
H = live steam heater.
E = exhaust steam heater.
P = circulating pump.
T = expansion tank.
A = condensation pipe.
G = grease extractor.
V = back-pressure valve.
M = engine.
C = condensation receiver.

systems, the "overhead" being advantageous where a large number of floors is served by one principal circuit.

Duplication of Pumps.—When pumps of the centrifugal type are used for circulating water, and their duplication is desired, the piping for joining them should be arranged in series owing to the characteristics of these appliances. A series connection, with both pumps operating at the same time, permits of the head being divided between them, and are more simply controlled than when operating in parallel. Fig. 51 shows how the piping for duplicate pumps may be arranged, with the necessary stop valves in position. For running in series the stop valves 5 and 6 are closed, whilst by the further adjustment of the valves, either the one or the other pump can be used alone.

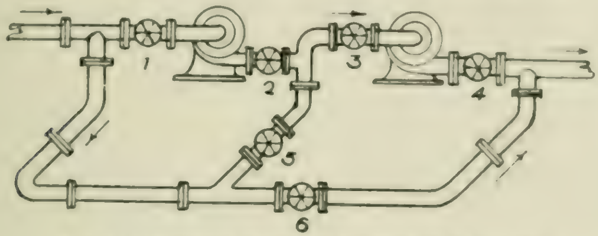


Fig. 51.—Showing valves for duplication of pumps.

Another form taken by a forced circulating system is shown in Fig. 52. This differs principally from the usual "two-pipe" arrangement in that a by-pass is provided at B between the flow and return risers, and in that the movement of water through the greater portion of the risers depends upon natural circulation. The principal object attained by this design is the utilization of water at a high temperature in the main circuits, whilst at the same time it is delivered to the heating surfaces at a much lower temperature. By circulating water at a higher temperature than is usually done, a smaller weight is required to produce a given heating effect, and the cost of operation is reduced. Into the base of each flow riser the highly-heated water is forced, this having its temperature reduced through mixing with the cooled return water in virtue of the by-pass B. The supply of heated water is controlled by a throttling device or regulated orifice at A, and so long as

it is correctly adjusted to suit the height of the risers and the requirements of the heating surfaces, the heated water will not make a short circuit with the main return. The risers when in pairs together with the by-pass make a complete circuit in themselves, whilst the positive and limited supply of the high temperature water serves the function of a separate heater. It will be conceived that in unit time every pair of risers will circulate a given weight of water, depending upon their height and the temperature drop allowed, and that if the volume

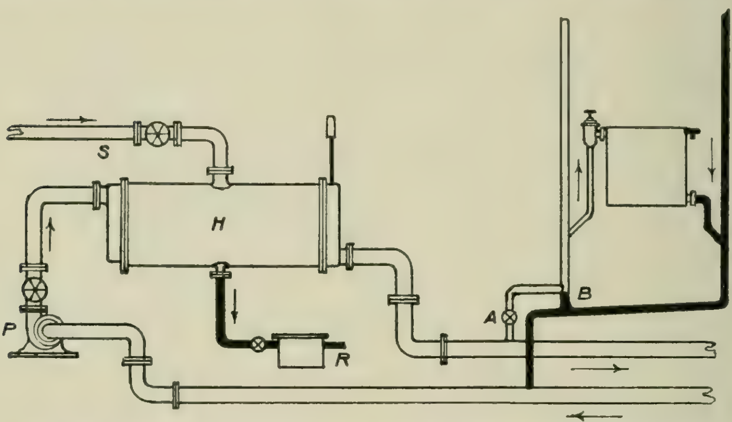


FIG. 52.—Forced circulating system with by-pass for risers as conceived by Captain Reck.

S = steam pipe.
H = steam heater.
P = circulating pump.
R = condensation outlet.
A = regulating device.
B = by-pass.

passing from A falls below that amount, the deficiency will be made good by the admission of the return water at B.

An example will aid in making the matter clear. Assume the risers in Fig. 52 are sized to circulate 20 lb. of water per minute for a temperature drop in the heating surfaces of 40° F., that is for a capacity of 40 × 20 = 800 B.Th.U. per minute. These same heat units would be supplied by 8 lb. of water falling through a temperature of 100° F. Now, if 8 lb. of water at the temperature of 240° F. were delivered per minute into the base of the flow riser, the circulation would proceed apace until the capacity of 20 lb. per minute were attained. The

temperature of the return water would be 240 – 100 = 140° F. and as 20 – 8 = 12 lb. of this would flow through the by-pass B, the temperature of the water in the flow riser would be reduced from 240 to an average of 180° F.

Methods of Regulating Temperature of Circulating Water.— The different methods of adjusting the water temperature to suit the changing weather conditions have been indicated in the figures considered, but where steam is the heating agent they may be briefly stated as follows:—

(1) The water temperature may be varied at the heater by increasing or decreasing the steam supply, by controlling the discharge of the water of condensation, and by regulating the pressure and degree of vacuum.

(2) The water at the heater may be maintained at a constant temperature, and the velocity of circulation varied.

(3) Both the temperature and circulating speeds may be altered conjointly.

The best practice to adopt, will depend upon the general design, Figs. 49 and 50 being best served by the first method given, whilst Figs. 48 and 52 are more suitable for the second and third methods of regulation.

When a heater, for example, as in Fig. 48 or 52, is supplied with live steam, the first method of regulation is not so good, owing to the power absorbed by the pump being the same under all conditions of the external air. In other words, no economy of operation would be shown when only the minimum heating effect is required. On the other hand, by curtailing the discharging capacity of the pump, the power to drive it may be substantially reduced. The extent to which economy can be carried by an electrically-driven centrifugal pump depends upon the size and type of motor, and the regulation adopted. As a general statement, when a pump is run at a fairly constant speed, the power to drive it may be diminished by about 25 per cent., if the discharging capacity is reduced by one half through the partial throttling of the outlet valve.

CHAPTER VII

LOW-PRESSURE "LIVE" STEAM HEATING SYSTEMS

Heat of Steam.—As a heating agent, steam is specially convenient, as it may be utilized for a large variety of purposes, both of an industrial and domestic nature.

The total heat of steam gradually increases with increase of pressure, but the latent heat value diminishes as the pressure increases. For example, at atmospheric pressure, the latent heat of steam is 970, and at 30 lb. gauge pressure 928 B.Th.U. per lb. In other words, each pound of steam during condensation at, and from atmospheric pressure will yield $970 - 928 = 42$ additional heat units to the condensation of the same weight at, the higher pressure. On the other hand, the total heat of steam per lb. at atmospheric pressure is 1150 B.Th.U., whilst that at 30 lb. per square inch is 1171 B.Th.U. Thus, more heat by 21 units is stored in the steam at the higher pressure; but it may not be available for heating purposes.

In this class of work it is usual to take the useful heat as equivalent to the latent heat value; but this holds good only when the condensation occurs at the same pressure as the steam. If, on the other hand, the condensation becomes subjected to a lower pressure, re-evaporation will occur; but heat will be lost, unless precautions are taken to prevent it.

Drop of Pressure in Steam Pipes.—To circulate steam through a system of piping a certain head or pressure is required, but the permissible velocity will be chiefly governed by the following considerations, viz. the flow of the steam and water whether in the same or in opposite directions, the height of the lowest heating surfaces above the boiler, also by the direct return, or otherwise, of the condensation to the boiler. As a rule, when a gravity

system is installed, only a small pressure drop should be allowed, and if this is observed there is less likelihood of the lower heating surfaces being flooded during the coldest weather.

Those unfamiliar with steam-heating work sometimes fail to realize what is meant by drop or fall of pressure. As already indicated, some force or energy is essential to cause steam to flow through pipes, and it is the absorption of this energy through friction that is responsible for the drop of pressure. For example, assume a circuit of a given length, where the boiler pressure is 5 lb. per square inch; if, by the time the steam has reached the far end of the circuit its pressure has fallen to 3 lb. per square inch, the drop of pressure would be 2 lb. per square inch. At the same time this differential pressure of 2 lb. per square inch would be counterbalanced at the boiler return, by the water rising in it to a height of $2{\cdot}4 \times 2 = 4{\cdot}8$ feet. It is the restoring of the equilibrium at the boiler return that is responsible for low-lying mains being sometimes flooded.

Water-Hammer in Steam Pipes.—It is important when arranging the pipes of steam-heating systems that no pockets are formed in which water will be retained, thereby interfering with the flow of steam. This may be avoided by adopting suitable fittings, by giving the pipes an adequate pitch, and by the use of drip pipes where the condensation tends to gather. Accumulations of water in steam pipes are detrimental and objectionable, in that they cause water-hammer or snapping sounds. Steam coming in contact with cold water, is rapidly condensed, when the vacuum that ensues is responsible for the projection of the pocketed water in the direction of the vacuum. The violence with which the water is precipitated against bends, tees, valves, or other fittings, is often the cause of their being damaged.

Clicking or snapping sounds usually arise through the steam coming in contact with smaller volumes of water, and, although these are not usually accompanied with any marked strain, they should be avoided as far as possible. A certain amount of snapping occurs when steam is readmitted into a system, for the cold extended surfaces bring about a rapid condensation, and produce a more or less differential pressure.

With a well-designed system, however, this is of short duration, the snapping ceasing when the pipes are heated.

Gravity Systems.—As the name implies, these are installations in which the water of condensation simply gravitates from the heating surfaces to the boilers, in contradistinction to those in which the condensation is returned by some external agency, such as pumps, injectors, or other lifting appliances. Generally speaking, gravity systems may be divided into two principal divisions, viz. low-pressure ones, or those using steam

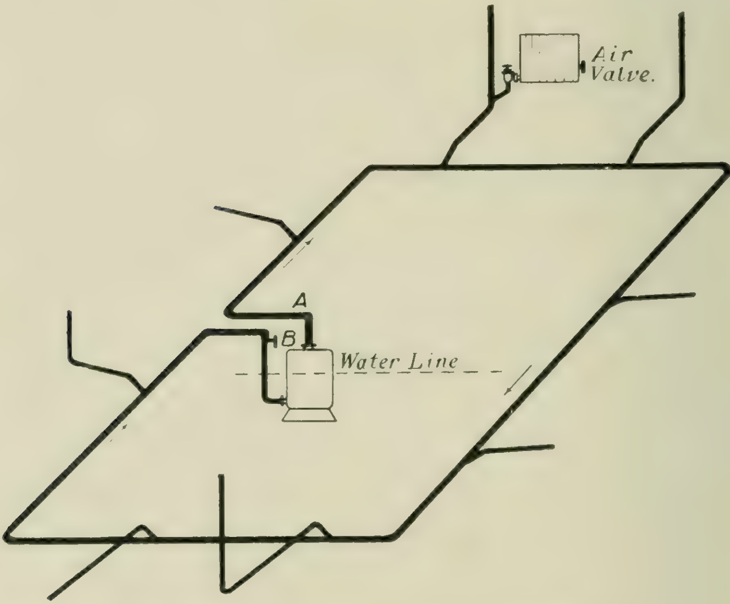


FIG. 53.—One-pipe system of steam heating.

above atmospheric pressure, and atmospheric systems, or those in which the steam falls within the heating surfaces to the pressure of the atmosphere.

The piping is arranged in different ways, and, as in low-pressure hot-water installations, may be on the up-feed or down-feed principle. Frequently, the terms "wet" and "dry" are used in connection with the returns, to indicate whether they are constantly charged with the water of condensation or not.

One-Pipe Systems.—In Fig. 53 a one-pipe circuit system is shown, and from the point A the main should have a fall of about 1 inch in 10 feet. This arrangement permits the steam and condensation to flow in the same direction in the main, thus avoiding conflict with these fluids. Excepting for the short vertical return that joins with the boiler, the main circuit should be of uniform bore, whilst the vertical part of the return may be reduced by one or two sizes. At the point B, an automatic

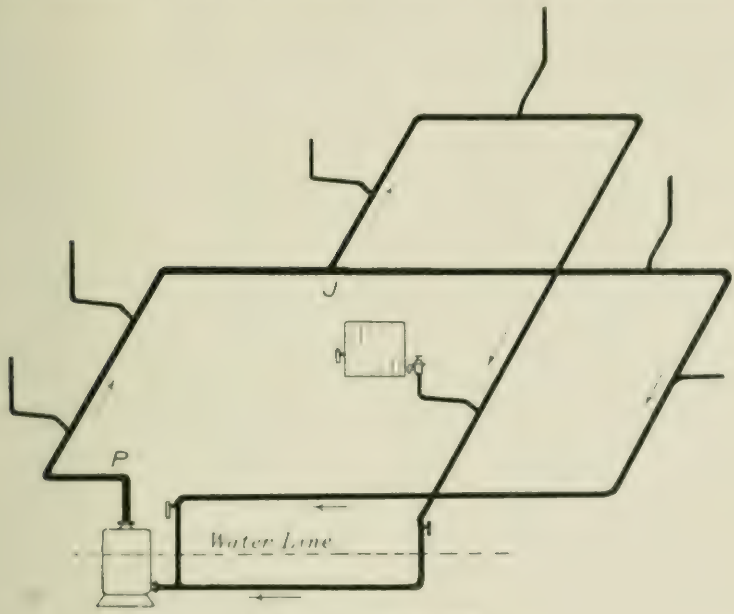


FIG. 54.—Divided circuit.

air valve is placed to free the circuit of air, for, unlike water systems, the air tends to seek the lower, instead of the higher level. Neither is the air so effectively dislodged as in water installations, for the density of air is only a little greater than that of the steam. For example, at atmospheric pressure, the density of steam is 0·037 lb. per cubic foot, whilst air at 212° F. weighs 0·059 lb. per cubic foot. With one-pipe systems, only one riser is required for supplying the heating surfaces, the steam and condensation passing through the same pipe.

Owing to the varied forms that buildings take, divided circuits are rendered necessary, these being indicated in Fig. 54. From the point J, two sections are formed, but the steam should follow a defined course, otherwise the drainage may be interrupted and the efficiency of a system impaired. There is no difficulty, however, in causing the steam to flow in one particular way, so long as the returns are joined below the boiler water-line. From point P, the mains are supposed to have a downward pitch, whilst an automatic relief for air is provided for each return.

The number of units into which a system should be divided will depend upon the circumstances of each particular case, and

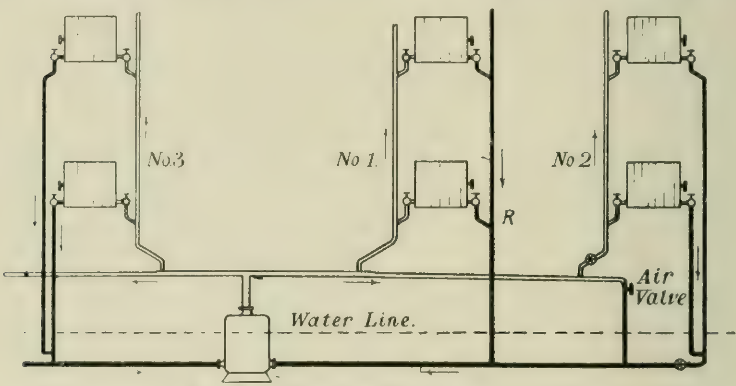


FIG. 55.—" Two-pipe " up-feed system.

the degree of control desired. For example, one floor may be provided with two or more circuits, each of which could be sub-divided into two or more parts, or one divided circuit may be used for two or more floors.

Two-Pipe Systems.—A " two-pipe " system of steam heating is given in Fig. 55. This differs principally from a " one-pipe " system, in that the steam mains are reduced in size as the heating surfaces are supplied; separate pipes are employed to handle the condensation, and the principal returns are charged with water. In the risers Nos. 2, and 3, it will be observed that a separate return is used for each radiator. This arrangement is adopted to make a system more silent in operation, for when two or more returns from radiators are joined, as in

risers No. 1, steam can enter the heating surfaces either through the inlet or outlet connections. The method indicated by riser No. 1, however, can be made to answer satisfactorily.

When diminishing the size of steam mains, eccentric fittings should be used, in order that true alignment may be preserved on the lower sides of the pipes. The common forms of diminishing fittings are unsuitable for steam pipes, for not only may they be the cause of snapping sounds, but the capacity of the mains may be much reduced.

With a two-pipe system, valves are attached to both the flow and return connections, and when stopcocks are introduced at the base of the risers, a simple arrangement for this is given on the right of Fig. 55.

Overhead or Down-feed Systems.—Fig. 56 gives an overhead system of piping, the steam from the boiler being directly conveyed to the overhead mains, whilst from these, the various drop-pipes are taken to supply the heating surfaces. The lower horizontal mains are arranged to remain charged with water, for it is essential that the steam should only gain admission to the drop-pipes from their higher ends. Although only one steam main from the boiler is shown, two or more may be used, the extra provision of course depending upon the size of the plant and the number of units into which it is proposed to divide it.

Radiator Connections.—The radiator connections of steam apparatus may be arranged in any convenient way, so long as they do not trap or retain the water of condensation. It is not essential to make any difference in the form of connection for the higher and lower heating surfaces, as is frequently done with hot-water apparatus, for the equal distribution of steam is more readily effected.

Obstructions to Pipe Lines.—It frequently happens when piping a building that some obstruction presents itself that necessitates a break or alteration in the alignment of the pipes. This may arise through the intervention of girders, beams, and the like, or through structural work requiring the raising of pipes from a lower to a higher level. Such irregularities in the piping often demand the use of "drips," and as these transmit the steam-pressure directly where they discharge into the returns, allowance must be made for this in sizing the pipes.

or the drainage from the more distant mains may be appreciably retarded.

In Fig. 57 two methods are shown for relieving the condensation, where, on the one hand, the steam main is passed beneath a girder, and on the other, where it is bent over it. Where, however, circumstances are favourable, the method at B

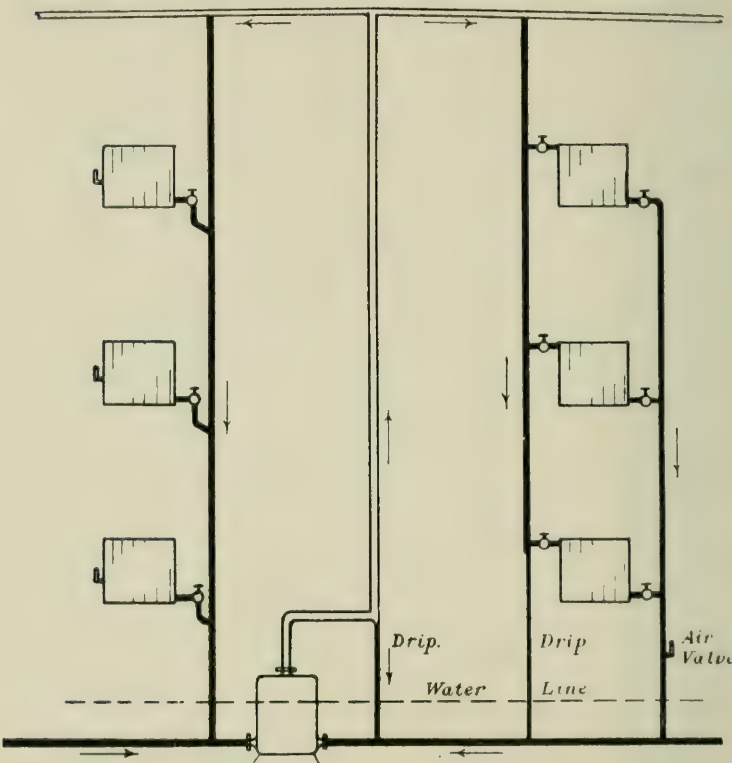


FIG. 56.—"Overhead" or down-feed system.

is the better one to adopt, for when "drip" pipes are used, suitable points must be found for their discharge.

False Water Lines.—Occasionally boilers are placed in positions that demand the use of false water-lines, in order to seal the ends of the returns. Fig. 58 shows what is meant, but in this case a difficulty is often experienced in maintaining the

"false" water-line, through a tendency to siphonage being brought about. An attempt is made to arrest this by the

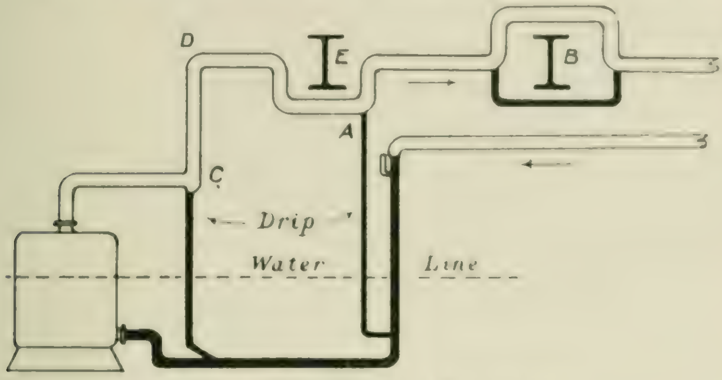


Fig. 57.—Showing methods of draining pipe.

provision of the equalizing pipe between the steam main and the top of the loop, but it does not fulfil effectively the purpose that is sought.

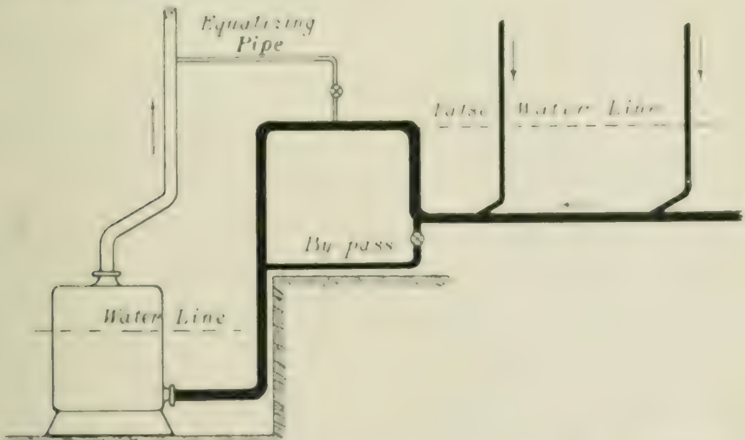


Fig. 58.—Common but defective method of making a false water-line.

The weakness arising in connection with the equalizing pipe of Fig. 58 is produced by the steam being brought intointimate

contact with the condensation at the loop through the oscillation that occurs. Condensation of the steam is brought about, which results in the partial vacuum removing the contents of the trapped return.

Another method of arranging the equalizing pipe for a "false" water-line is shown in Fig. 59, where it is joined with a main return some 30 feet or so distant from the loop. The purpose of the long equalizing pipe is to enable a small volume of air to be trapped so that it may form a cushion between the steam and the condensation, whilst at the same time the necessary pressure is transmitted to the loop. This method, however.

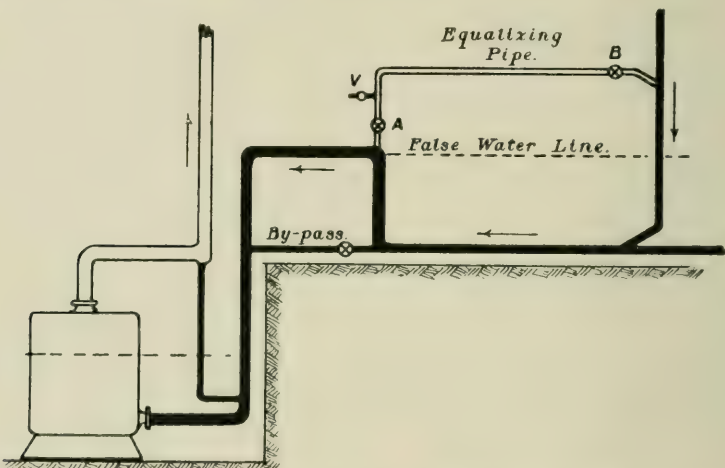


FIG. 59.—False water-line provided with long equalizing pipe.

may not be proof against siphonage, but it will prolong the intervals between the actions, and is better than the practice shown in Fig. 58. To charge the equalizing pipe with air, water must first accumulate in the loop when the stopcocks A and B are closed. In a short time, the steam in the equalizing pipe is condensed, when, upon the opening of the valve V, air is readily admitted to supply the partial vacuum that is formed. The air valve V is now closed, and the valves A and B partially opened, so as to admit the steam pressure being gradually transmitted to the return.

A better method of forming a "false" water-line is shown in Fig. 60, in which a constant discharge trap is employed. This method is specially suitable for large apparatus, and where the return water is subject to a pronounced cooling action. The water of condensation, it will be seen, enters at the bottom of the trap, in order that its water surface may be kept as steady as possible. As in Fig. 59, the equalizing pipe is joined with a return riser, which is well removed from the trap, and the connection is made so as not to drain the water of condensation from the return. To put the trap into action, the water is

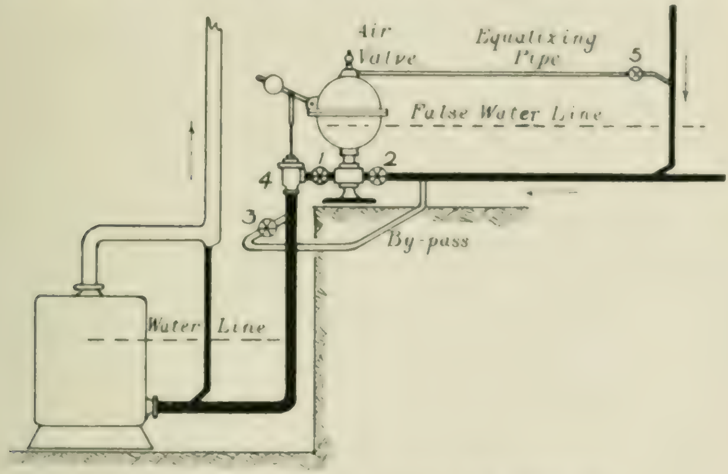


FIG. 60.—Better method of making false water-lines.

allowed to rise to its normal height, when the valves 1, 2, and 5 are closed, and No. 3 on the by-pass partly opened. The steam in the equalizing pipe and trap is given time to condense, when the valve on the trap is opened to admit air. This being done, the air valve is closed as well as No. 3, whilst 1 and 2 are opened wide; finally, valve 5 on the equalizing pipe is partially opened when the steam pressure is admitted to the trap.

Dry Returns. "Two-pipe" Systems.—In the preceding pages on steam heating, the importance of sealing the returns has been pointed out where a silent working and efficient system is desired. It is not convenient, however, nor yet practicable in

every case to introduce low-lying wet returns, and when these cannot be utilized it is advisable to arrange and size the piping, that the drop of pressure may facilitate the drainage of the condensation from the different branches.

Figs. 61 and 62 show two methods of treating the returns from four coils, and will aid in the elucidation of the point under consideration. Let it be assumed that each coil in Fig. 61

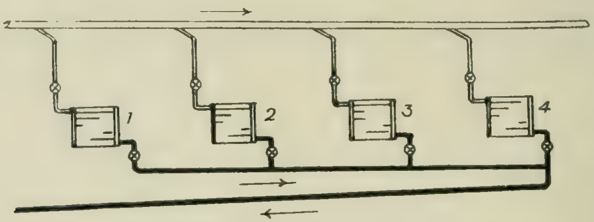


FIG. 61.—Method of treating return piping.

is of the same size, and that the pressure of the steam falls at any given but equal rate. This being granted, it will be clear that the steam pressure at the inlet of coil No. 1 will exceed by a certain amount that at the inlet of coil No. 4; similarly, the pressure in the return of Fig. 61 will be greater at the outlet of coil No. 1 than that of No. 4. The effect of this arrangement

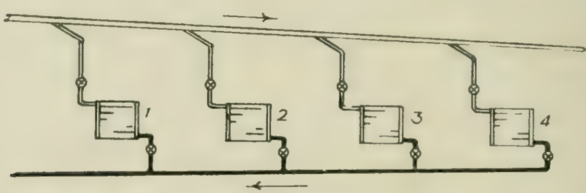


FIG. 62.—Defective arrangement of return piping.

is to cause the surplus pressure at the first coil to drive the condensation in the direction desired.

On the other hand, when the returns are treated as in Fig. 62, the pressures at the outlet of the coils are the same as in Fig. 61, but the surplus pressure at the first coil holds back the condensation from the others until the hydraulic pressure is sufficient to overcome the resistance imposed. More or less

noise may be caused by the water surging backwards and forwards, whilst the heating capacity of the coils beyond the first would be somewhat impaired.

Cases where Special Appliances are required for Returning the Water of Condensation.—When possible, an apparatus should be erected so that the condensation can gravitate to the boilers, but in some cases this cannot be done, either because it is not practicable to place the boilers low enough, or because steam is used which requires to have its pressure reduced. Under the circumstances first named, it may only be essential to artificially return the condensation from the lowest surfaces,

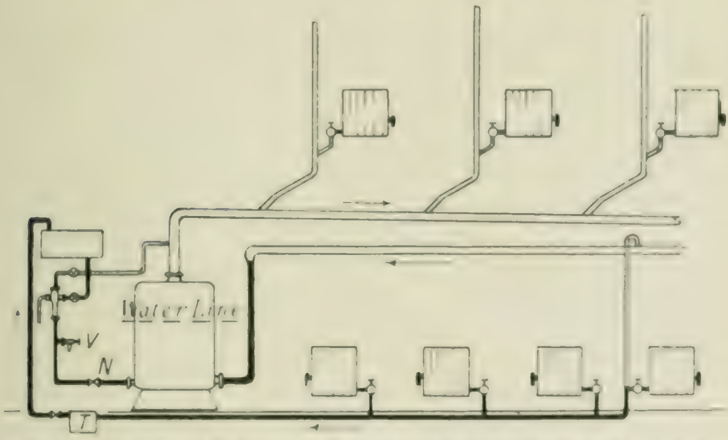


FIG. 63.—A method of handling the condensation from low-level radiators.

whilst in the other case, the whole of the condensation would require to be handled. Occasionally, the condensation may be passed to waste, but where this is done an automatic feed is necessary to replace the water that is lost. In all systems a certain volume of make-up water is essential to replace the leakage at valves and other fittings, but in a well-arranged gravity system, this loss should only be small.

A simple and comparatively cheap appliance for returning the condensation is an injector, and although its application is limited in low-pressure work, it will answer very well where the heating surfaces are not extensive, and where the water-line

of the boiler is frequently observed. In Fig. 63 a case is given where an injector is employed for returning the condensation to the boiler from a few radiators at a low level, whilst from the higher heating surfaces the condensation is returned by gravitation. From the steam trap T, the water is displaced by the terminal pressure of the steam, and with any suitable trap the water should be elevated about two feet for each lb. per square inch of pressure. Thus, if the piping is sized for a drop of pressure of $\frac{1}{4}$ lb. per square inch, and the condensation requires to be raised through a height of 6 feet, the boiler pressure should not be less than $\frac{6}{2} + \frac{1}{4} = 3\frac{1}{4}$ lb. per square inch. The fittings that accompany a low-pressure injector are indicated in Fig. 63, V being a relief cock that requires to be opened when starting the appliance.

Other appliances for handling the condensation are shown in the following chapter.

General Remarks.—With respect to the merits and drawbacks of "one-" and of "two"-pipe systems, it is sometimes contended that the latter is the less noisy of the two, but this depends very much upon how the systems are designed. With a one-pipe system, larger pipes are necessary, owing to the steam and condensation travelling through the same channels; but this is not always a serious drawback, as the capacity of the pipes increases rapidly with increase of size. The chief advantage of two-pipe systems is that smaller pipes can be used, although this is more than off-set by the extra length of piping required and by the use of additional valves.

A failing more or less common to all low-pressure systems of steam heating is that the valves must be either turned fully on or shut off. If a radiator valve were partly open in the case of a one-pipe system, the passage through it would be insufficient to admit steam and drain away the condensation. The result of this would be to hold up the water in the radiator until its pressure was sufficient to overcome that of the steam, whilst with the ordinary automatic air relief, the leakage of the condensation may be brought about.

The extent to which the failing occurs in two-pipe systems depends very much upon the height of the heating surfaces, and upon the treatment of the return connections. If,

for example, the radiator connections are made as at R (Fig. 55), the closing of the inlet valve when the return one is left open would have the same effect as the single connection already described. On the other hand, if the inlet valve of a radiator on, say, No. 2 riser of Fig. 55, were closed, and the return valve left open, differential pressure would arise through the condensation of the steam, when water would rise in the return until a height was reached to restore equilibrium. Under ordinary circumstances the differential pressure produced in this way may be considered as nearly equal to the initial pressure of the steam. As an approximate guide it may be taken that each pound of differential pressure per square inch will raise the condensation through a height of 2 feet 5 inches.

For the most economical operation of a low-pressure plant it is necessary that the water of condensation be returned to the boiler at as high a temperature as possible, and at little or no cost. Return traps provide a simple automatic means of handling the condensation from low situations where the terminal pressure is suitable. The principal cost of this method is in the initial outlay for the appliances, the operating and maintenance charges being comparatively small.

CHAPTER VIII

FITTINGS FOR LOW-PRESSURE STEAM SYSTEMS

Pressure Reducing Valves.—These appliances take various forms, the type to use depending upon the pressure to be carried in the heating system, the extent to which the pressure must be lowered, and the degree of sensitivity desired. In heating plants where very low pressures are necessary, some form of diaphragm reducing valve is usually adopted, as this form enables a large differential pressure to be maintained between the inlet and outlet sides.

Fig. 64.—Pressure reducing valve by Kieley and Mueller.

Fig. 64 shows a form of reducing valve for realizing on its outlet side a pressure as low as 1 lb. per square inch. It is simple in construction, the valve of the equilibrium type being closed when the outlet pressure acting on the upper side of the diaphragm exceeds the upward effort due to the weight and lever. On the other hand, when the pressure at the outlet tends to fall, the lever exerts the superior force, and the valve is opened to the flow of steam. This form, however, is only suitable for installations when the delivery is not subject to rapid fluctuations.

Another pattern of reducing valve is given in Fig. 65, which is intended for situations where the pressure on the outlet side exceeds 5 lbs. per square inch, in other words, where a very

sensitive valve is not imperative. At a point a few feet distant from the valve and on the low-pressure side a small pipe is taken, the other end being joined with the diaphragm chamber at the top of the valve. A stopcock should be pro-

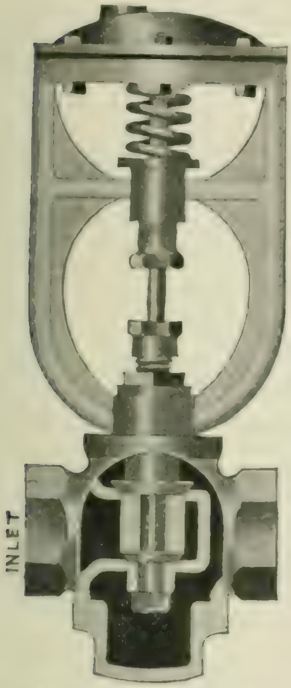


FIG. 65.—Pressure reducing valve. By Kieley and Mueller.

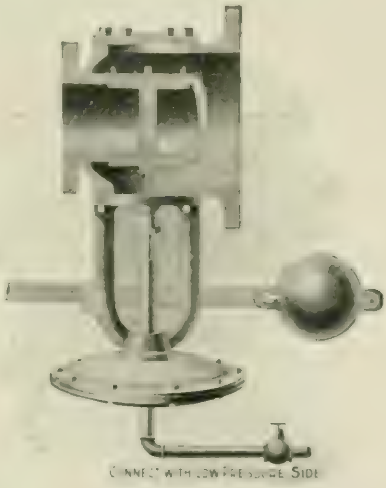


FIG. 66.—Reducing valves for low pressures. By Kieley and Mueller.

vided on this pipe, the throttling of which reduces pulsation, when a rapid change of pressure is brought about.

A construction giving a more sensitive form of reducing valve is shown in Fig. 66. Like the previous one, the diaphragm is independent of the valves, this being advantageous in that it can be joined with the low-pressure side some distance away, so as to respond to the average pressure of a system. The main valve is of the equilibrium form, the diaphragm being large enough to give any low pressure desired, whilst

any tendency to pulsation can be avoided by throttling the small pipe that communicates with the diaphragm.

Many reducing valves are very troublesome, so that every care should be taken to select a type suitable for the work in hand.

Air Valves.—The small difference between the densities of air and steam makes it a difficult matter to know the best point

FIG. 67.—Automatic air valve for steam.

for locating air relief valves on ordinary low-pressure systems; for, instead of the air accumulating at one particular place, it tends to diffuse over a large area. So far as heating surfaces are concerned, the air valves are usually placed from one-third to two-thirds their height on the side opposite to the steam supply, whilst on the mains they are located at low points.

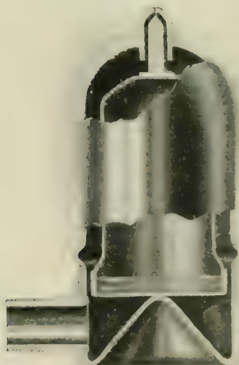


FIG. 68.—Automatic air valve. By National Radiator Co.

For the relief of air, automatic valves are universally adopted, and, although they differ in form, they mostly depend for their action upon the expansion and contraction principle. They are not, however, an ideal means for affording relief, but as yet there is nothing better to take their place.

A well-known automatic air valve is illustrated in Fig. 67, which operates by the expansion and contraction of a composition plug, C. Like all similar contrivances, it depends upon a decrease of temperature accompanying the accumulation of air, under which circumstances the plug

contracts and the air escapes. For adjusting the valve, the screw B is used, this being slackened when the steam is first turned on, and being gradually screwed up until none escapes.

Another form of air valve is given in Fig. 21, p. 43, in which a float is used. Briefly explained, its action depends upon the expansion and contraction of the air confined in the annular space, which communicates with the float chamber by means of a small aperture near the bottom of the fitting. Upon the expansion of the air, the water from the annular space is displaced into the float compartment, the float is rendered buoyant and so the air outlet is closed. On the other hand, the contraction of the air causes the water to leave the float chamber, and, in consequence, the float falls.

Still another type is indicated by Fig. 68. In this, a metallic vessel is used which contains a volatile fluid that is readily vaporized with heat. When in the gaseous state, pressure is exerted within the vessel, and as end deflection brings about its elongation, the outlet orifice is closed. The gathering of air, however, in the vicinity of the valve causes a cooling action to set in, when the vapour reassumes its fluid form, and the internal pressure is removed.

Radiator Valves.—These are usually the same as for low-pressure water heating, and are considered in an earlier chapter.

Steam Traps.—In low-pressure heating plants, steam traps are often necessary, in the first place, to receive and regulate the discharge of the condensation, and secondly to prevent waste of steam. These appliances take a variety of forms, but it is bad economy to procure a type in which the initial cost is the chief recommendation, for the loss through steam leakage may soon more than offset the difference in cost between the inferior and superior appliance.

Fig. 69 gives a steam trap of the box type, where the valve, by means of a quick screw motion, is opened and closed by the falling and rising of the float. The interior of the trap is not subjected to the pressure of the steam, and, unless discharging against a head of water, may be open to the atmosphere. In construction, the float valve differs from the usual form in that the condensation is discharged through the flo[illegible]

itself. When the trap is not in use, or when the condensation is flowing into it, the float falls, owing to the water entering it. If, however, the steam reaches the float, the water from the latter is dislodged, and in being rendered buoyant the valve is closed. After a short period, the steam in the float condenses, the water re-enters and causes it to fall, when a further

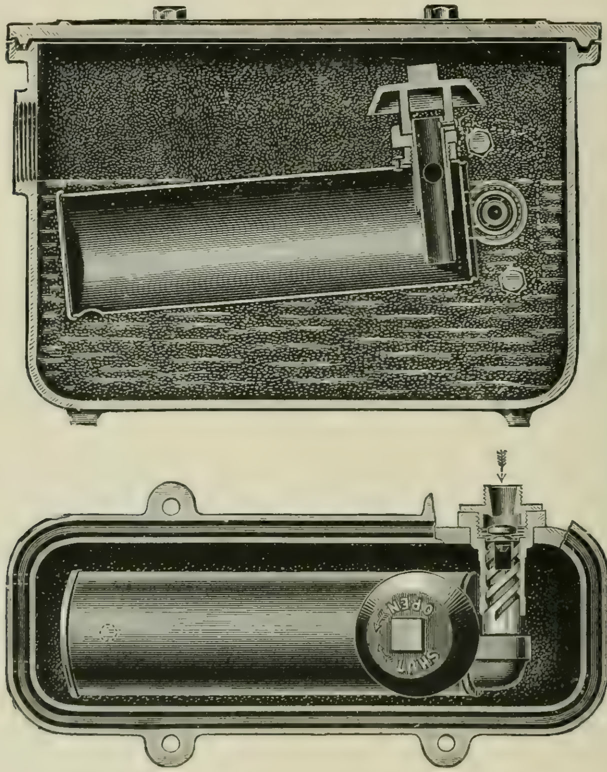


Fig. 69.—Steam trap. By Lancaster and Tongue.

discharge of condensation is effected, if any has accumulated at the inlet of the trap. On the other hand, if in the interval no condensation has occurred, the steam immediately reappears and brings about the buoyancy of the float. For the satisfactory working of the trap it is necessary that air should escape from the float, and for this purpose an adjustable air

valve is provided. The air valve also serves the purpose of preventing irregular action through the re-evaporation of the condensation when it enters the float at a high temperature.

Another box trap is shown in Fig. 70. In this case, a bucket float that is pivoted to the casing on the right is used, the motion opening and closing a double-seated valve. The inlet is not shown, but it is usually located on one side of the outlet, the trap in this case being subjected to the full pressure

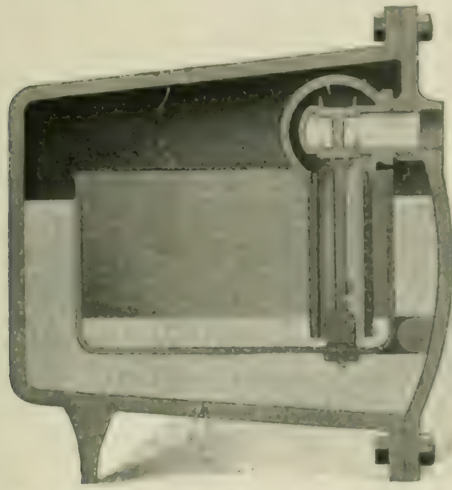


FIG. 70.—Steam trap: low-pressure type. By Kieley and Mueller.

of the steam. Upon the condensation being delivered into the trap, it overflows into the bucket, which, when nearly full, sinks by virtue of its weight, and so opens the valve. As the interior of the trap is subjected to the steam pressure, the water is displaced until the float is buoyed up by the surrounding water, and the valve is again closed. The form of valve in Fig. 70 is only suitable for low pressure, a single valve with restricted orifice being generally adopted for higher pressures. For the removal of air, a hand-controlled valve is provided.

Where smaller traps are necessary, expansion forms are largely used; but many of these are faulty through lack of sensitivity of the expanding parts.

Fig. 71 gives a trap of the expanding type, the valve being

closed when steam comes in contact with the inner tube. Here it is a case of differential expansion, the inner tube expanding in a greater degree than the body of the trap. In the absence of steam, the trap is open to the air, and the condensation drains freely away. In order to permit of the necessary movement of the valve, the trap should be of moderate length, the overall length for a $\frac{3}{4}$-in. size, being 2 ft. 10 in., whilst the external diameter is $1\frac{1}{2}$ inch.

Another type of expansion trap is shown in Fig. 72, and although its external form is similar to the previous one, its construction is entirely different. To impart a pronounced motion to the valve, a very expansive material is employed in the inner tube H, the action of the trap depending upon the differential expansion between this substance and the inner tube, and not that between the inner and outer tubes, as in Fig 71. When steam enters the trap, the composition within the tube is readily heated, and the expansion that results, forces along the piston D, which in turn, causes the tube to move bodily in the direction of the inlet, and so the valve is closed. This particular action may not be apparent at the outset, but when it is taken into account that the spring at C, is considerably stronger than that at J, and that the movement of the tube must be in the direction of the smaller resistance, the action will be the more readily conceived. Upon the condensation gathering in the trap, the tube begins to cool, and as the medium inside contracts, the piston D is forced back (owing to the spring at J, and the pressure of the steam), the valve is again opened, and the condensation discharged. The purpose of the spring at C is for setting the trap to suit the pressure of the steam, and also for preventing the trap being subjected to excessive strain.

When selecting a steam trap in which a sensitive expanding medium is employed, it is important to observe that this does not prove a source of weakness. All steam traps possess some drawback, but by carefully observing their construction, and knowing the requirements to be fulfilled, there should be no difficulty in selecting a trap of a suitable type.

Return Traps.—For automatically returning the water of condensation from low positions, return traps are often employed.

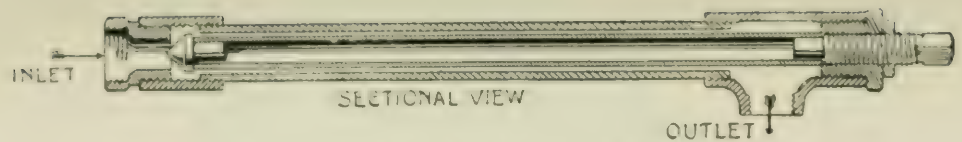


Fig. 71.—Steam trap. By Engineering Specialities Co.

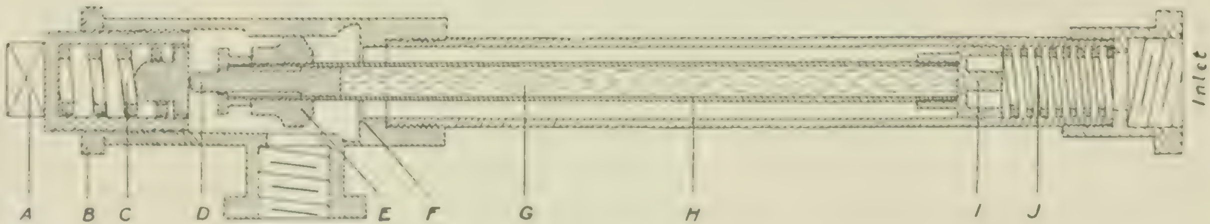


Fig. 72.—The "Horne" steam trap.

A = adjusting screw. B = lock nut. C = strong spring. D = piston. E = face of valve. F = valve seating. G = expansive material. H = tube. I = guides. J = spring.

These appliances vary in structural details, but the underlying principle is usually the same.

One form of return trap is given in Fig. 73. The various return pipes discharge into a receiving tank, which is located at the lowest point, and from this point the condensation is raised to the trap, which should be located not less than 3 feet above the boiler water-line. As the water accumulates in the return trap, a float is buoyed up, which opens a valve by means of which steam is admitted directly from the

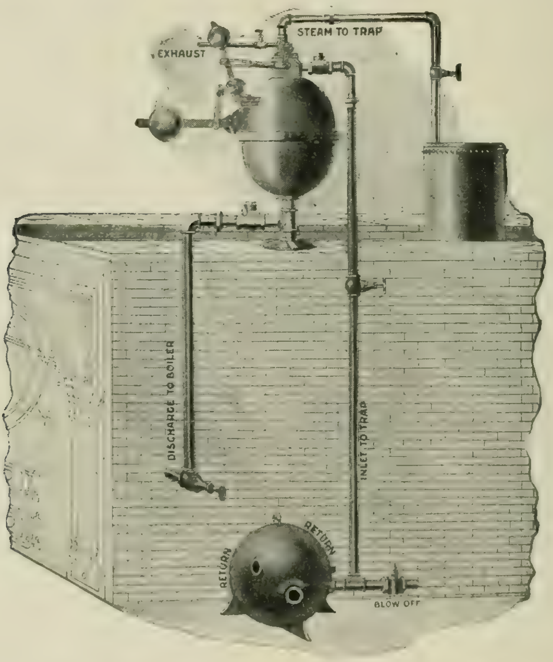


FIG. 73.—Return trap. By Kieley and Mueller.

boiler. The effect now produced is the equalization of the boiler pressure on the inlet and outlet sides of the trap, when, on account of its position, the contained water is able to gravitate to the boiler. As the condensation is being discharged the float begins to fall, and when a given level is reached (governed by the slack motion of the lever) the steam supply is cut off. If at the end of the discharge, the return trap is not opened to

the air, a partial vacuum will be created through the condensation of its contained steam, which will assist in the elevation of the water, along with the terminal pressure of the steam.

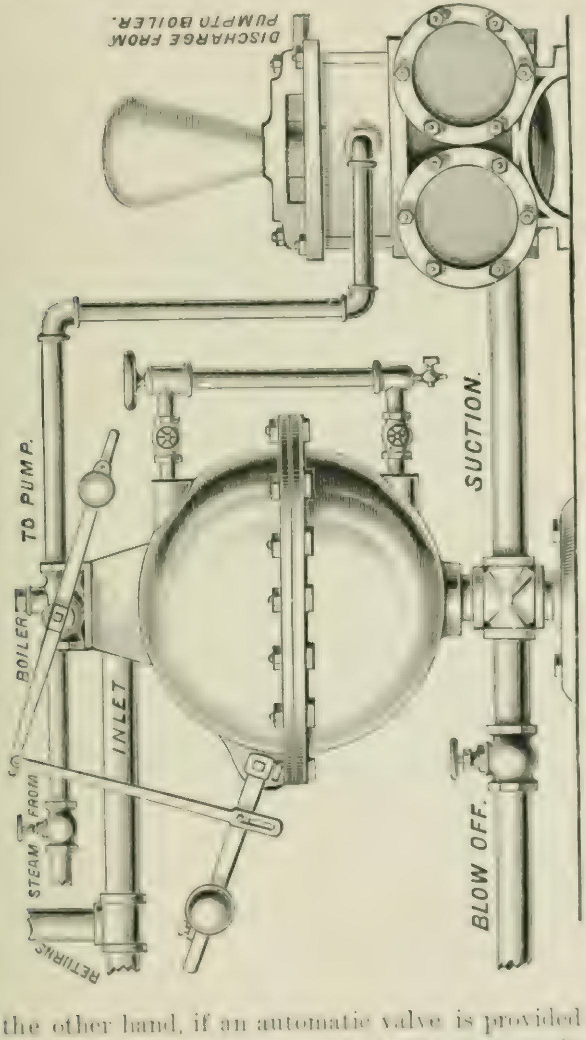


Fig. 74. Pump and receiver. By Kieley and Mueller.

On the other hand, if an automatic valve is provided that will admit air when the steam supply to the trap is shut off, the raising of the condensation will depend entirely upon the pressure at the receiver.

With regard to the general aspects of return traps, in which a partial vacuum is, or is not used, each method has its own merits and limitations. The chief advantage arising from the formation of a partial vacuum in a return trap, is, that for any given case the condensation can be raised through a greater height than where the steam pressure of the container is solely relied upon. Any slight leakage, however, will soon destroy a

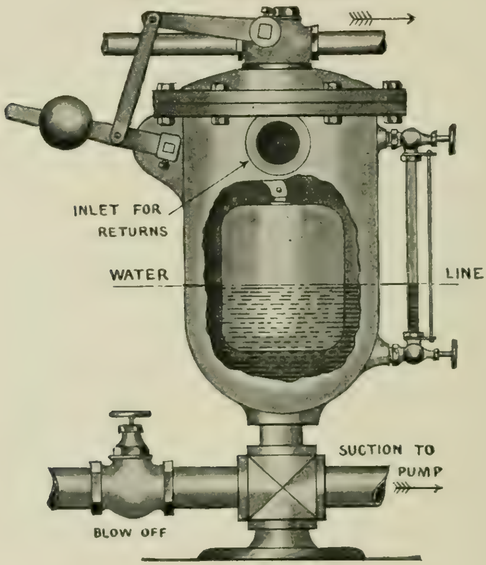


FIG. 75.—Receiver showing float.

vacuum, and if the elevation of the condensation depends mainly upon it, the trap may be somewhat slow and uncertain in its action. There is no difficulty, however, in bringing about a rapid condensation in the return trap, and increasing the degree of vacuum, through the introduction of a water spray, but it has the drawback of further complicating the appliance.

Pump Receivers.—In works and large places, where power boilers are used, the latter are often utilized for supplying the steam to the heating system. In low-pressure heating, this necessitates the steam pressure being reduced, with the result

that the water of condensation requires to be handled by some special appliance. As already shown, this can be done by the use of return traps, but for returning large volumes of condensation, pump receivers are more suitable.

In Fig. 74 a general view of a pump receiver is shown where

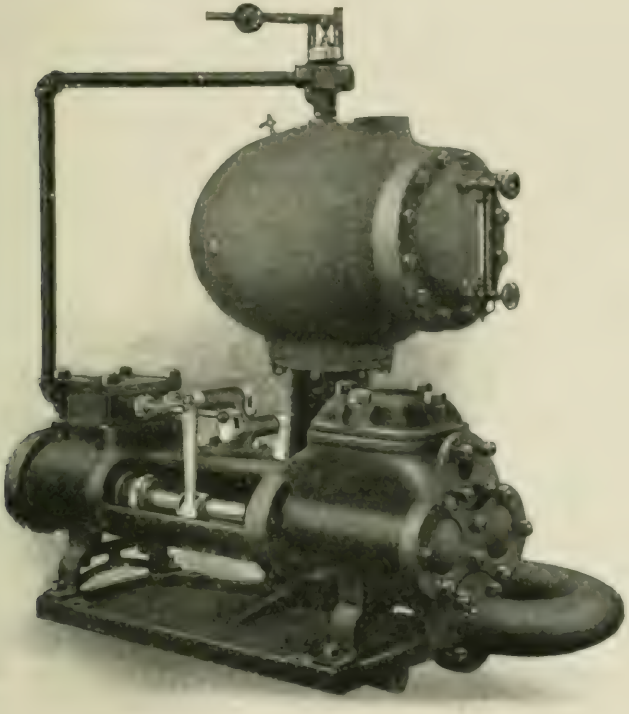


FIG. 76.—Pump and receiver. By Worthington Pump Co.

the condensation is assumed to gravitate to it. For its action, the appliance depends upon a float in the receiver which operates the valve by which steam is admitted to the pump. Thus the speed of the pump is automatically adjusted to deal with the varying rates of condensation, and the latter may be either directly returned to the boiler, or delivered to any other point.

In Fig. 75 the returns are indicated as discharging into the upper part of the receiver, but where "wet" returns are used it is often more convenient to join them at its base. To prevent the receiver under the latter conditions being subject to differential pressure and so interfering with the water-line, an equalizing pipe is joined to the upper part of the receiver.

Another pump-receiver is given in Fig. 76, and although it is similar in action to the previous one it differs in details of construction. In the receiver of Fig. 76 the float employed is of the bucket type, this being attached to one end of the lever, whilst a counter-weight is secured to the other end. The float is filled with water, but upon the condensation rising in the receiver to a predetermined height, the float is rendered buoyant by virtue of the counter-weight, the steam valve opening and putting the pump into action. The removal of the water beyond a given point in the receiver is accompanied by the opposite movement of the float, and the steam supply to the pump is curtailed.

As a rule the automatic valves used in connection with receiver pumps do not entirely cut off the steam supply, for so long as a portion of a heating system is in use, the minimum condensation at any period will be sufficient for the pump to "creep." This is also advantageous in that the pump is more suitably maintained for speeding up at any moment desired.

CHAPTER IX

EXPANSION OF PIPES

DEFECTS frequently arise in a system of piping through inadequate provision being made for its expansion, for where free movement is prevented, sufficient stress may be caused to distort a pipe permanently, and its fracture may be only a matter of a short time. In some cases, where more or less provision is made, defects appear, for unless a pipe is arranged to expand in a particular direction and from specific points, considerable strain may be concentrated in the wrong place.

Provision for expansion may be made by arranging the pipes that they can be sprung, by means of expansion joints and special bends, and in some cases by subjecting the pipes to a tensile strain when they are jointed.

The Springing of pipes may be resorted to at branches and at changes of direction, where the pipes are free to move, but the extent of the movement requires to be regulated by the safe permissible strain.

Example 7.—A wrought-iron branch is taken from a main pipe as in Fig. 77, this being of 3-inch bore and firmly anchored at B, whilst the main pipe is secured at A. Let it be assumed that the pipes expand in the direction of the darts, that the distance between Ax is 60 feet at 40° F., and that the steam pressure carried is 10 lb. per square inch (equivalent temperature 240° F.). Determine (a) how much the pipe L expands when heated from 40° to 240° F.; (b) the distance the anchor B should be placed from x in order that no damage is done through the strain set up.

The increase of length due to expansion may be found by the formula—

$$r = 12Lct \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where r = expansion in inches,
l_e = length of pipe in feet,
c = coefficient of linear expansion,
t = rise of temperature in degrees F.

To ascertain the minimum length of the strained portion l of Fig. 77, the following formula may be used :—

$$l = 0{\cdot}36\sqrt{\frac{\mathrm{E}d\mathrm{L}ct}{f}} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$\text{or } l = k\sqrt{d\mathrm{L}t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

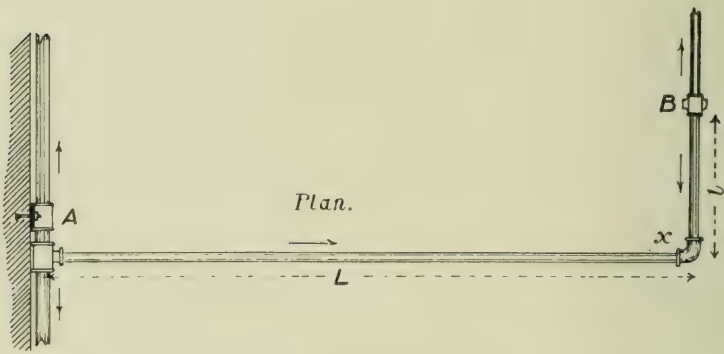


Fig. 77.

where l = minimum length of strained or deflected pipe,
L = length of piping causing deflection in pipe l,
E = modulus of elasticity in lb. per sq. inch,
d = external diameter in inches of deflected pipe,
f = safe working stress of metal in lb. per square inch,
$k = 0{\cdot}36\sqrt{\frac{c\mathrm{E}}{f}}$
c and t are the same as for formula 4.

TABLE III.

Properties of Metals.

Metal.	Linear coefficient of expansion. c.	Modulus of elasticity lb. per sq. in. E.	Safe working stress in lb. per sq. in. f.	Value of K.
Cast iron	0·0000062	18,000,000	4,500	0·057
Wrought iron	0·0000068	26,000,000	12,000	0·043

Using formula 4 for ascertaining the expansion for the case given, we have—

$$x = 12lct$$

Substituting values $x = 12 \times 60 \times 0.0000068 \times (240 - 40)$
when $x = 0.98$, **or nearly 1 inch.**

For the second part of the problem, using formula 6

$$l = k\sqrt{d L x}$$

Assume the external diameter as $3\frac{1}{2}$ inches, whilst k is taken from Table III.

Substituting values, $l = 0.043\sqrt{3.5 \times 60 \times 200}$

$$l = 0.043 \times 204.9$$

when $l = 8.8$, **or say 8 ft. 10 in.**

Thus, for Example 7, the point of anchorage B should be about 9 feet removed from the bend at x, whilst this length

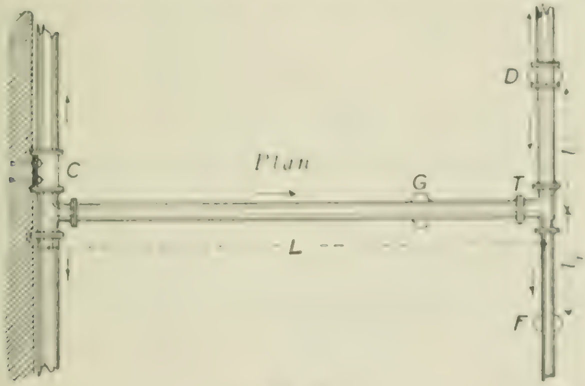


FIG. 78.

would be deflected about one inch through the expansion of the pipe L. Although in Fig. 77 no other fixings are indicated than at A and B, it will be understood that roller or other suitable supports would be necessary at regular distances apart.

Example 8.—Determine the distances for the points of anchorage from the branch T for a case as represented by Fig. 78, where the pipes are firmly secured at C and D. The

pipes are of cast iron, the external diameter of L being 6 inches, that in the direction of D 5 inches diameter, and that towards F 4 inches diameter. Assume the pipes convey steam at 5 lb. pressure per square inch (equivalent temperature 228° F.), whilst the length L at 48° F. is 30 feet.

For this example the minimum length l between D and T is determined by the aid of formula 6, where

$$l = k\sqrt{d\mathrm{L}t}.$$

The value of k in Table III. is given as 0·057.

Substituting values, $l = 0{\cdot}057\sqrt{5 \times 30 \times (228 - 48)}$

$$l = 0{\cdot}057 \times 164$$

when $\boldsymbol{l = 9{\cdot}35}$**, or say 9 ft. 4 in.**

If now a fixing is used at F which prevents lateral, but permits of longitudinal, movement, the distance from T to F may be obtained as above. Or, as the length will vary directly as the square root of the diameter when the remaining conditions remain unaltered, it may be found by proportion, when

$$l_1 = \frac{9{\cdot}35 \times \sqrt{4}}{\sqrt{5}}$$

$\boldsymbol{l_1 = 8{\cdot}36}$**, or say 8 ft. 4 in.**

Thus, for the arrangement in Fig. 78 and the conditions given, the anchorage D should be approximately 9 feet 4 inches from T, whilst the support F in the line should be about 8 feet 4 inches from the branch. The fixing at G, if permitting of lateral motion, may be located close to the branch, but if it only allows movement lengthwise its correct position may be ascertained by formula 6.

Expansion Joints.—In straight lengths of pipes where ample provision for expansion cannot be made by bends and the straining of pipes, expansion joints are generally used. For these appliances to work satisfactorily, they should be securely fixed, whilst the pipes themselves must be arranged to move in the direction of the expansion joints. This is readily done by anchoring the pipes at given points, and by providing roller or other movable fixings.

There are two principal types of expansion joints. The first is provided with a sleeve-piece, which slides through a "stuffing" box, whilst the second depends upon a flexible diaphragm or disc, the maximum strain upon which should be kept well within the safe elastic limit of the metal.

In Fig. 79 is shown a common form of sliding joint for pipes of small bore, whilst a joint for larger pipes is given in Fig. 80. To the latter, lugs are attached in order that it may be readily secured in position.

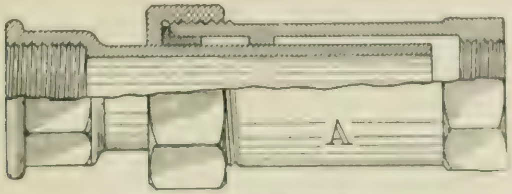


FIG. 79.—Expansion joint.

The chief weakness of sliding joints is their tendency to leak, either through the failure of the packing material or through the straining of the movable ends when the alignment is not perfect. To preserve a true alignment between the expansion joints and piping, different practices are adopted, such as the fixing of permanent guides to the joints, or by using ball and socket joints.

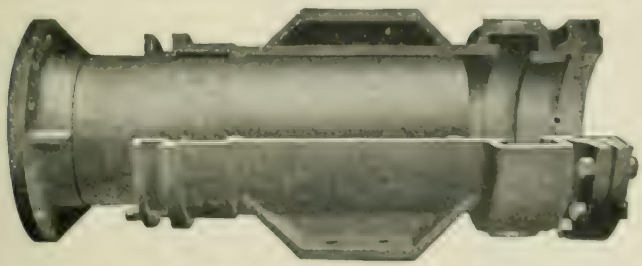


FIG. 80.—Webster expansion joint.

Their principal merit is in the length of movement they allow, but when fixing, care should be observed that the sleeves are well withdrawn before the pipes are attached.

Fig. 81 gives a disc or diaphragm expansion joint of the double type, expansion being provided for on both sides of the fitting. In construction, the body part consists of an inner and two outer cast-iron rings, whilst between these rings the outer edges of two copper diaphragms are clamped. The inner edges

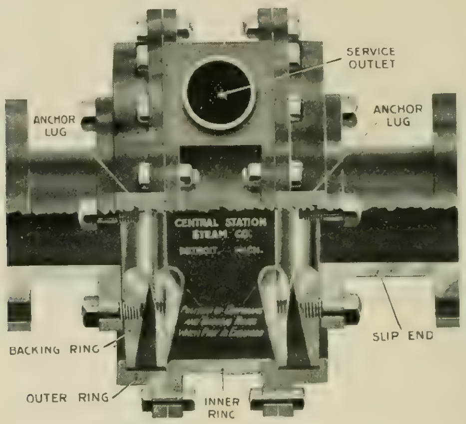


FIG. 81.—Diaphragm expansion joint.

of the diaphragms are passed through and spun over cast-iron "backing" rings, which limit the movement of the diaphragms and form a suitable attachment for the "slip" ends of the joints. In the figure, it will be observed that the outer and inner cast-iron rings are arranged to form rebates, in which the movement of the "backing" rings is confined.

Another form of diaphragm joint is shown in Fig. 82. In this, the movement is effected on each side of the appliance, one edge of a corrugated copper disc being connected with the end where the pipe is joined, whilst the other edge is secured between the large cast-iron flanges.

The diaphragm class of expansion joint has no stuffing box, and is therefore very suitable for positions difficult of access. For this reason, they have been largely used on underground pipes for conveying both water and steam.

Generally speaking, the double diaphragm joint provides expansion for 100 feet of pipe, whilst large sliding joints will

take up the expansion from at least 300 feet of pipe. Taking the unit of length as 100 feet and the unit temperature range as 100° F., the expansion of wrought- and cast-iron pipes will be 0·82 inch and 0·75 inch respectively. Thus, if the maximum

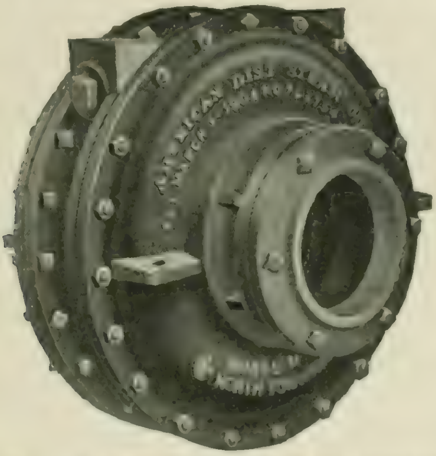


FIG. 82.—Diaphragm expansion joint or "variator."

temperature range in a system of wrought steam piping is 200° F., the expansion to be provided for each 100 feet should be not less than $0{\cdot}82 \times 2 = 1{\cdot}64$ in.

Expansion Bends.—Where space permits, expansion bends

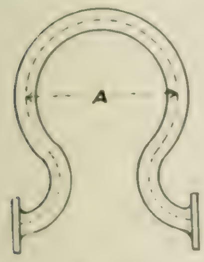


FIG. 83.—Expansion bend.

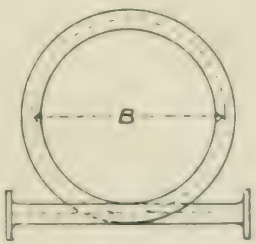


FIG. 84.—Expansion bend.

are frequently used, two common forms being shown in Figs. 83 and 84. For these, copper instead of iron tubing should be adopted, as the latter is more liable to failure through the rigidity of the material, especially when the ends are screwed.

For either copper or iron expansion bends, flange joints are preferable, as, when ordinary screwed joints are used, any weakness introduced by the threads permits the whole strain to be concentrated at such points. When the ends of wrought-iron pipe are screwed into flanges such joints may be strengthened by welding. A few years ago the welding of such joints was not such a practicable thing as at the present time, but with the advent of the oxy-acetylene welding apparatus it can be simply and economically carried out. Moreover, when flanges are welded to pipes no previous threading need be done, the ends being slipped simply through the flanges and the blowpipe applied.

Should expansion bends be required on vertical pipes they may take the form of a helix.

The following table gives the principal dimensions of the expansion bends shown in Figs. 83 and 84.

TABLE IV.

Dimensions of Expansion Bends.

Internal diameter in ins.	Dimensions.			
	A.		B.	
	ft.	ins.	ft.	ins.
2	1	6	2	6
$2\frac{1}{2}$	2	0	3	0
3	2	4	3	9
4	2	9	4	9
5	3	0	5	3
6	3	6	5	6
7	3	9	6	0
8	4	0	6	6
9	4	6	7	0

Application of Tensile Strain during Jointing.—This method consists of fixing pipes rather short, so that when two ends are drawn together they are subjected to a tensile strain. Upon the application of heat, however, expansion begins, and the tensile strain diminishes, falling to "zero" when a certain temperature is reached. If, however, the temperature continues to rise, the pipes in turn are subjected to a compressive strain, provided they are unable to move in a

longitudinal direction. Fig. 85 will help to illustrate what is meant.

Example 9.—If a wrought-iron pipe 40 feet long is rigidly held at C and D (Fig. 85), determine the temperature range through which it may be heated without overstraining it. Also

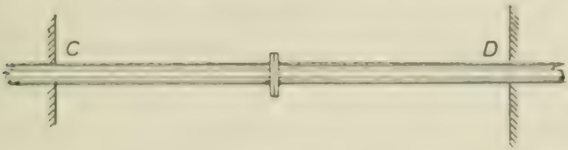


FIG. 85.

give the distance the flanges should be apart prior to the pipe being jointed.

If the temperature range is calculated from the neutral point or that of no strain, it may be obtained by the following formula:—

$$t_n = \frac{f}{c\mathrm{E}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where t_n = temperature from neutral point,
f = maximum stress allowed per square inch,
c = coefficient of linear expansion,
E = modulus of elasticity in lb. per sq. in.

For the values of f, c, and E see Table III., which, for the case under consideration, will be 12,000, 0·0000068, and 26,000,000 respectively.

By formula 7—

$$t_n = \frac{f}{c\mathrm{E}}$$

and substituting values, $t_n = \dfrac{12{,}000}{0{\cdot}0000068 \times 26{,}000{,}000}$

when $t_n = \mathbf{68^\circ}$ **F.**

The distance between the flanges should be equal to the expansion from the neutral point, and is obtained by formula 4, where

$$x = 12\,l_1 c t.$$

The value of t here will be 68 as obtained above. Substituting values

$$r = 12 \times 40 \times 0{\cdot}0000068 \times 68,$$
$$\text{when } r = \mathbf{0{\cdot}22}, \textbf{ or, say, } \tfrac{1}{5} \textbf{ inch.}$$

For the example given, the maximum temperature through which the pipe could be heated without exceeding the safe stress used would be $68 \times 2 =$ **136° F.**

If the flanges were closer or further apart prior to jointing, and the pipes were heated through the same range of temperature, the strain would exceed 12,000 lb. per square inch.

Tensile strain, it will be seen, is limited in application, but there are cases where it can be advantageously adopted, and especially when the pipes can be sprung a little from a bend at the same time.

CHAPTER X

ATMOSPHERIC SYSTEMS OF STEAM HEATING

THE term "atmospheric" is used in connection with installations where the pressure falls to that of the atmosphere upon steam entering the heating surfaces. In other words, atmospheric systems are those in which the radiators or other heating surfaces and the return mains are open to the air.

When compared with ordinary low-pressure apparatus, atmospheric systems of steam heating have much in their favour, for they eliminate most of the drawbacks of the former whilst their installation is comparatively cheap. In the earlier atmospheric systems the boiler pressure carried was usually about 2 lb. per square inch, whilst with more modern apparatus it is limited to a few ounces per square inch. Moreover, much progress has been made in the design of the valves for regulating the steam supply, whilst the failings of the earlier forms have been overcome to a great extent.

A simple "atmospheric" system is given in Fig. 86, the boiler fittings being omitted. The water of condensation is drained to the receiver R, from the top of which an air-pipe is taken and joined with the boiler chimney, the draught of which is utilized to free the pipes from air. The receiver R is graduated in ounces per square inch, to indicate the boiler pressure, the water column being observed by a gauge glass.

When pressure is generated in the boiler the water from the latter is dislodged to the receiver, until the rising water column exerts the same pressure as the steam. To prevent the boiler plates being burned by too great a displacement of water, a steam relief is provided which comes into operation when a given pressure is exceeded.

It is the usual practice in these systems to join the steam

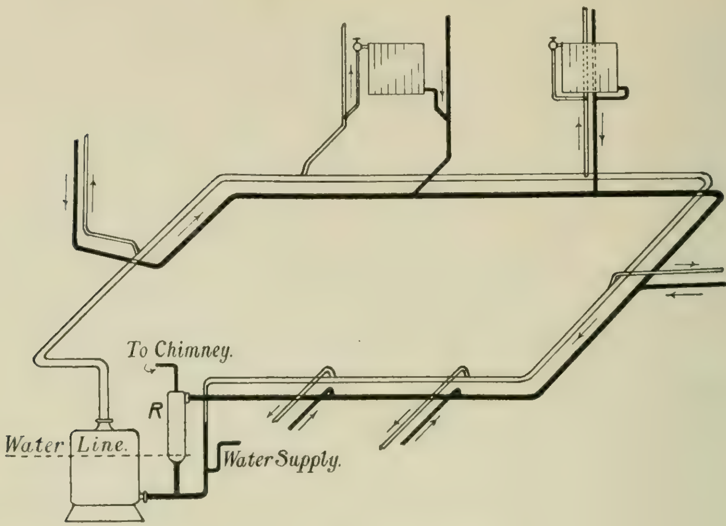


FIG. 86.—Atmospheric system.

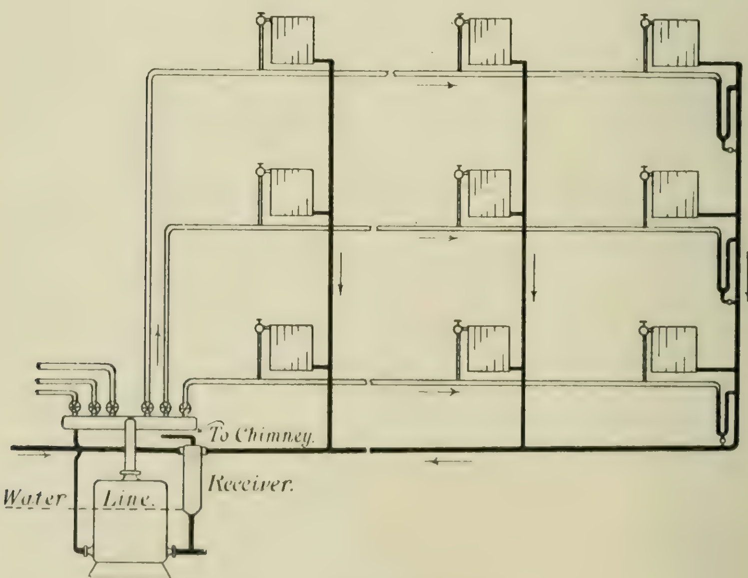


FIG. 87.—Atmospheric system.

supply to the highest parts of the heating surfaces, for by so doing, the air is the more effectively removed. No air valves are used, these being unnecessary with open returns.

The piping of "atmospheric" installations is much simpler than that of ordinary systems, and they are not subject to water hammer and similar sounds if carefully installed. For dividing into any given number of units, these systems are specially

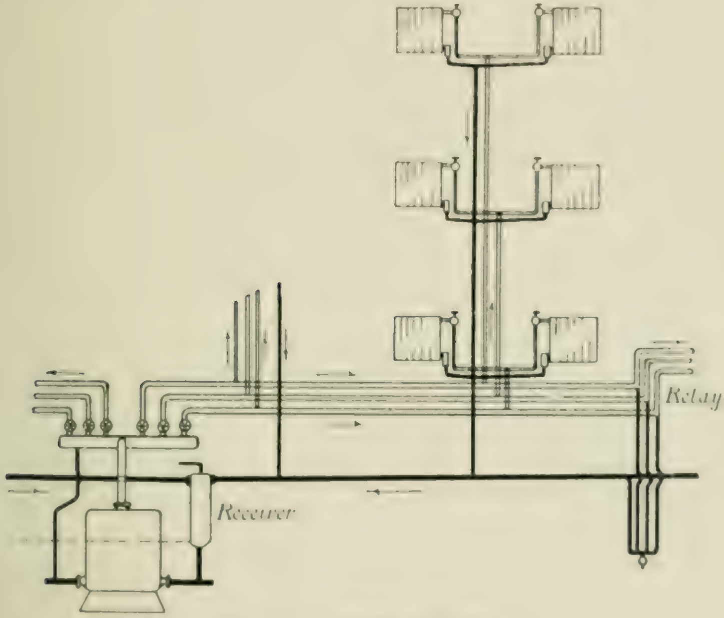


FIG. 88.—Atmospheric system.

advantageous, as the condensation from any radiator may be discharged into the most convenient return.

A piping system for large buildings is shown in Fig. 87. In this case, separate mains are used for each of the different floors, whilst any unit may be used independently of the remaining ones. The ends of the steam mains are trapped before being joined with the returns, the depth of these being adequate to resist the greatest steam pressure to be carried.

Fig. 88 gives another method of piping for large buildings. Here the steam mains are run from the "header" along a wall

or on a basement ceiling, separate risers and mains being used to serve the radiators on the different floors. Where mains are prolonged, "relays" are often necessary, either through some obstruction intervening in the line of piping or through the mains having fallen to too low a level. These, however, present no difficulty so long as trapped connections are made between the lowest points of the steam mains and the returns.

In Fig. 88 the steam risers and returns are connected with

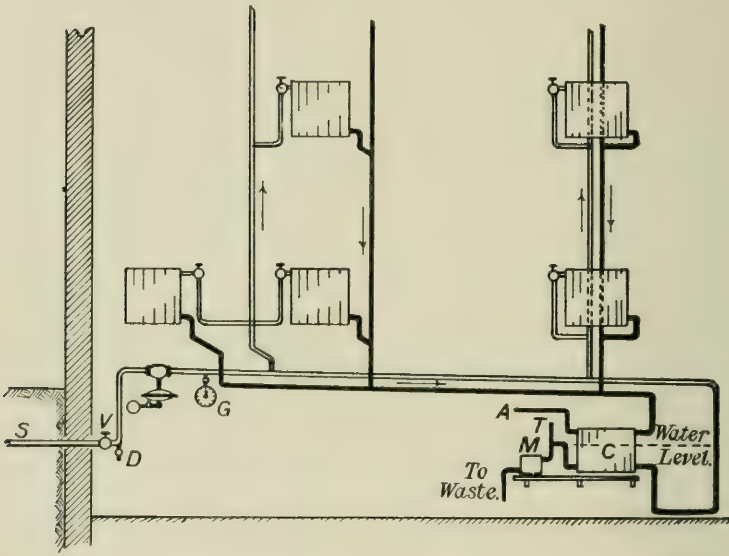


FIG. 89.—Atmospheric system where steam is supplied from an external source.

one side of the radiators, and this arrangement answers satisfactorily for radiators of a moderate size. For large heating surfaces where the connections are desired as shown, a distributing pipe may be used inside them.

Fig. 89 gives a general arrangement of the piping and fittings used, where the steam supply is obtained from a district main. The pressure of the steam in the service pipe S may be anything between 1 lb. and 5 lb. per square inch, this depending upon the distance from the district heating station. After entering a building the steam pressure is reduced to the desired

degree by the pressure-reducing valve shown. A cooling and condensing radiator is located at C, which serves to condense any steam that may reach that point, and to deprive the condensation of a large percentage of its heat before it is passed to waste. In general, no attempt is made in district heating to return the condensation, for unless the conditions are specially favourable, it is found much cheaper to waste it. From the cooling radiator the water of condensation passes through a meter, M, which records the weight of steam condensed, whilst the drip pipe from the steam main may join the cooling radiator in the manner shown.

Regulation of Atmospheric Systems.—From an economical standpoint, the success of these systems chiefly depends on the use of suitable appliances to prevent the waste of steam.

For radiator and other heating surfaces there are two modes of regulation. The first depends upon the use of "fractional" radiator valves, which are designed to admit varying volumes of steam according to the condition of the weather, whilst plates and pointers are frequently attached to indicate the extent to which they are opened. As one size of valve is used to serve varying amounts of heating surface, the valve or the piping should be arranged that only a limited volume of steam can enter a radiator, even when the valve is widely opened. In most cases, the steam supply is restricted, so that for a given external temperature, the heating surfaces cannot be filled to within 10 to 20 per cent. of their full capacity, the valves being partially closed as the weather gets milder, or as the condensing capacity of the surfaces diminishes.

The prevention of waste by this form of regulation depends upon the radiator valve being properly manipulated by the occupants of a room, or by the use of thermostatic contrivances, any failure in either respect allowing steam to flow directly into the returns. On the other hand, if due attention is paid to the temperature of a room, or the thermostatic appliances remain in order, not only is waste avoided, but the temperature of the condensation as it leaves the radiators is very appreciably reduced.

In the other mode of regulation, both the inlets and outlets of heating surfaces are governed, fractional valves, as before, being used for the steam supply, whilst some thermostatic or

fixed device is connected with the outlets. The principal advantage gained in this case is the reduced waste of steam.

A fractional radiator valve is shown in Fig. 90. The design of fractional valves should be such that the wire-drawing effect of the entering steam will not cut the valves and seatings, whilst the wear on the restricting orifices should be as nearly uniform as possible.

Fig. 91 gives a fitting for the outlet regulation of radiators.

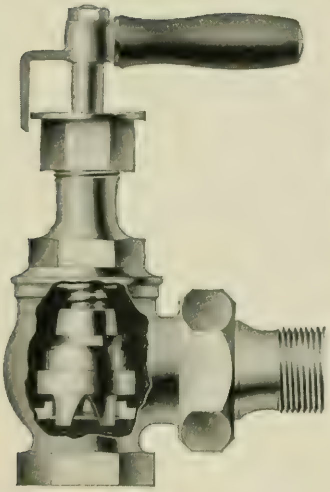


Fig. 90.—Jenkins "fractional" valve.

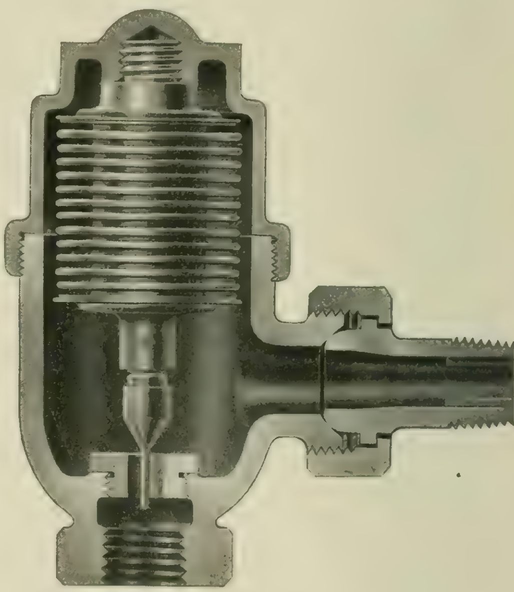


Fig. 91.—Webster Sylphon thermotor.

As already indicated steam should not appear at the outlets of atmospheric systems when the regulation is correct, although this is likely to occur, to a more or less extent, in all systems dependent upon hand control. The opening and closing of the valve are effected by the contraction and expansion of the corrugated vessel, which is rendered the more sensitive by the volatile fluid it contains. So long as steam is absent the valve orifice is open, and providing for the escape of both condensation

and air. If, however, steam comes in contact with the metal vessel the valve is closed.

The boiler draught of these systems also requires to be under accurate control in order that the pressures may be adjusted to meet the varying demands for steam. In Fig. 92 one method of doing this is shown, although special forms of

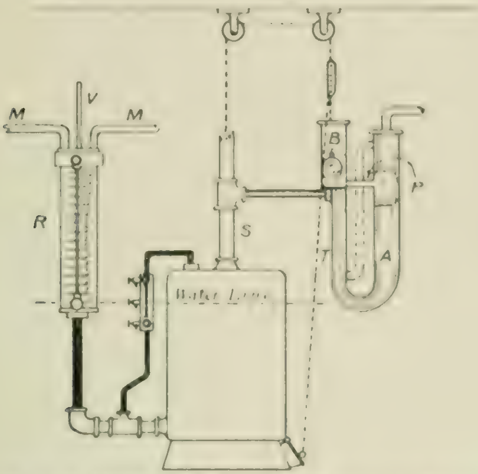


FIG. 92. Boiler of atmospheric system with automatic draught control.

diaphragms or other means are used. To a branch from the steam main S, the regulator and relief tube are fixed, the depth of the water seal being proportional to the blow-off pressure desired. In the upper tube B, a float is placed which is connected with the dampers of the boiler, whilst the level of the water in the float tube is influenced by the pressure of the steam on account of the cross connection P. As the actual steam pressure is represented by the vertical distance between the water-line of boiler, and that of the receiver R, it is imperative when fixing the latter that the zero point corresponds with the boiler water-line. The relief of steam when the pressure is excessive is effected through the smaller pipe that is joined at T as soon as the water-level is depressed to that level.

When steam is taken from an external source, as in Fig. 89,

the water of condensation is sometimes discharged to a pipe receiver instead of to a condensing radiator as shown. Some central heating companies, however, require condensing radiators to be installed, for where meters are used there is less likelihood of steam passing away unrecorded when it has once entered a system. Fig. 93 gives a pipe receiver discharging into a condensation meter.

The "Broomell" System of vapour heating is an atmospheric

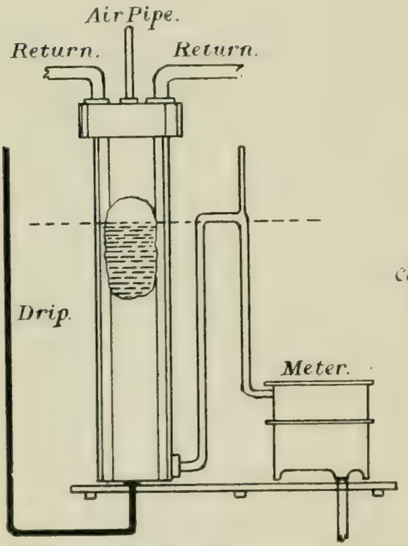


FIG. 93.—Pipe receiver and meter.

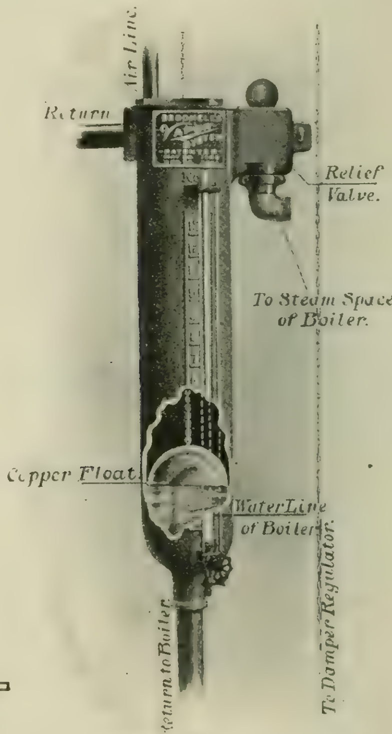


FIG. 94.—Broomell's receiver.

one, the apparatus being installed on the lines laid down, although some of the fittings differ in construction from those already shown. For this apparatus the receiver is shown in Figs. 94 and 95, but it performs two other functions as well, viz. it regulates the boiler draught by the float arrangement

shown, and it affords relief when the pressure of the steam rises too high.

It will be observed in Fig. 95 that the return pipes enter a water seal, the air pipe from this going to a condensing coil which is located above the receiver. From the coil the air pipe is joined with the nearest flue. In the same figure the

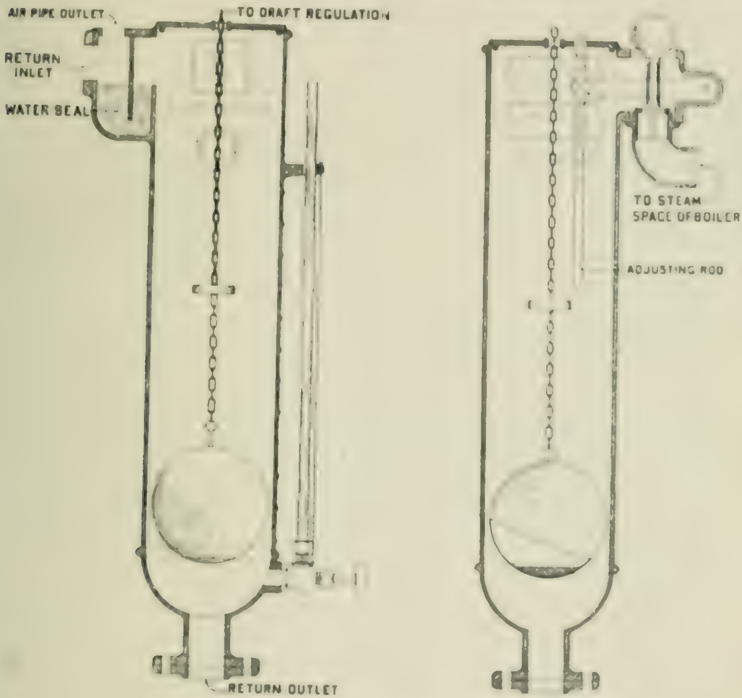


FIG. 95.—Broomell's receiver.

steam release valve is shown, which is opened by the raising of the adjustable rod when the float is buoyed up to it.

Fig. 96 gives the radiator valve that is used in the "Broomell" apparatus, this being made in six different sizes for attaching to ¾-inch supply pipes. From the illustration, it will be seen that the seating of the valve contains four ports, all of which are uncovered when the valve is opened wide. As less steam is required, one or more of the ports will be closed, the

degree of regulation being indicated by the relative position of the lever handle.

The special outlet fitting that is used is shown in Fig. 97. This appliance is attached to the outlet of each radiator or other

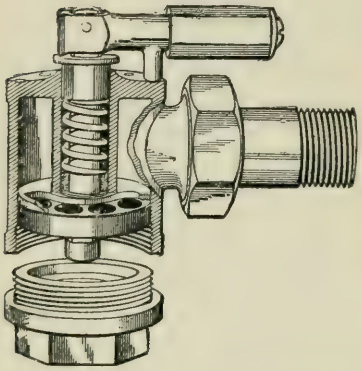


FIG. 96.—Broomell's radiator valve.

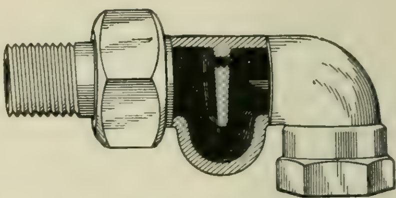


FIG. 97.—Broomell's radiator outlet connection.

surface, the small trap serving to break the full bore passage so far as the steam is concerned between the heating surfaces and the return pipes. At the same time the condensation can readily escape, whilst air or a little steam passes through the aperture in the lip.

CHAPTER XI

EXHAUST STEAM HEATING

ALTHOUGH "exhaust" heating is frequently combined with vacuum systems, this and the following chapter are confined to the utilization and value of exhaust steam, in ordinary "low pressure" and in "atmospheric" systems.

In works and other buildings where steam power is required, considerable economy may be often effected by combining the heating installation with the power plant employed. Before "exhaust" steam, however, can be delivered from the engines into a system of piping, some back pressure must be necessarily put upon them, and this has the effect of reducing their efficiency, owing to the mean effective pressure on the pistons being lowered. With a well-designed and suitable size of heating plant, the loss of efficiency of non-condensing engines will be very small, and may not realize 5 per cent. where 100 per cent. is taken as the basis when operating under normal conditions.

Generally speaking, it is economical to place back pressure on an engine for heating buildings when the heat required exceeds that available from the extra steam consumed by the engine in virtue of the added back pressure. To exhaust into a heating system, less than 2 lb. per square inch of pressure can be used (depending upon the general design), but whilst a case may be simple to solve where non-condensing engines are in use, the problem becomes more involved if engines of the condensing type are installed. An example at this point will aid in the explanation of the above so far as non-condensing engines are concerned.

Example 10.—At a given speed a non-condensing engine develops 80 horse-power, and uses 30 lb. of steam per horse-

power-hour when exhausting to the external air. If it is found that the steam consumption is increased by 10 per cent. when exhausting into a heating system, determine the smallest effective area of heating surface that will justify the increased consumption, also the advantage gained when the whole of the steam can be condensed in the heating plant. Let it be assumed that each square foot of heating surface transmits 250 B.Th.U. per hour, that each pound of exhaust steam yields 850 B.Th.U., the same weight of high-pressure steam 990 B.Th.U., and that 8 lb. of water are evaporated per pound of fuel consumed.

In the first place the extra steam consumed by the engine when exhausting into the heating system is $\frac{80 \times 30 \times 10}{100}$ = 240 lb. per hour. This weight of high-pressure steam will serve $\frac{240 \times 990}{250}$ = 950 square feet of heating surface. For a heating area less than the above, it would be more economical to blow the exhaust to waste, and to take the steam for heating directly from the high-pressure boiler. The smallest effective heating surface should therefore exceed 950 square feet.

In the second place, the coal to be charged against the heating account when exhaust steam is used, is that required to generate the extra steam owing to the back pressure that is carried. In this case the additional fuel consumed for a period of 1000 hours will be $\frac{1000 \times 240}{8}$ = 30,000 lb. or 13·35 tons. Before the weight of the fuel for live steam heating can be found, the area of the heating surface that will condense the whole of the "exhaust" must be ascertained. The latter is done by multiplying the total steam consumpt by 850, and dividing the result by 250. The total steam consumpt will be $(80 \times 30) + 10$ per cent. = 2640 lb. per hour, whilst as exhaust steam it will serve $\frac{2640 \times 850}{250}$ = 8976 square feet of heating surface.

Supposing now that the latter amount of surface were supplied by steam from a heating boiler at about 2 lb. per square inch gauge pressure (latent heat value 966), the fuel

consumpt in 1000 hours would be $\frac{8976 \times 250 \times 1000}{966 \times 8}$ $= 290{,}372$ lb. or 129·63 tons; that is, when taking the rates of evaporation of the two different boilers as equal. A saving is thus shown in favour of exhaust heating for the conditions given of $129·63 - 13·35 = 115·28$ tons of fuel, which represents a substantial cash value.

In general, it is not often that the "exhaust" from large steam engines can be wholly utilized for warming purposes, and although different conditions will give different results, the example taken shows where the efficiency begins, and to what extent it may be carried.

Heat in Exhaust Steam.—During recent times, controversy has been rife upon the relative merits of "exhaust" and of "live" steam for heating work. Cases have been cited in which it is claimed that greater economy in the consumption of fuel has been obtained by running a power plant for the sake of the exhaust steam, rather than taking high-pressure steam direct from the boilers. Now, so far as the "useful" heat of steam is concerned, it is clear that for any unit weight that enters an engine, the heat value of the "exhaust" is diminished by a certain amount, so that, if for a particular case, "exhaust" steam appears to give a better heating effect than "live" steam, the cause lies in some other reason than that pertaining to the available heat from either kind. The chief reason, no doubt, for the said superiority of "exhaust" steam when used for heating buildings, is that it gives up its heat more readily than live steam, especially when the latter is suddenly reduced from a very high to a low pressure. In other words, the moist condition of exhaust steam renders the heating surfaces more effective. One reason for this is, that when steam has its pressure reduced, it is superheated in passing through the reducing valve, and as this tends to dry the surfaces of the pipes and radiators their heat transmission value is diminished. Much will depend, however, upon the distance the steam is conveyed; but where an installation, as a whole, is fairly compact, the actions will be somewhat as described. Thus it will be apparent, that in order to give a certain heating effect by two different qualities of steam, the heating surface

in the one case will require to be more liberal than in the other.

At any given pressure, the total heat of "live" and of "exhaust" steam is the same, provided that they both have the same degree of saturation. "Exhaust" steam, however, usually contains entrained moisture, which lowers its quality from a thermodynamic point of view, but this proves advantageous where its rapid condensation is desired. On the other hand, where "exhaust" steam requires to be transmitted a considerable distance, a large percentage of free moisture is a drawback, for excessive condensation may occur in the pipes. Under such circumstances as these, some superheating is advantageous, either through a good quality of "exhaust" being suddenly dropped from a high to a low pressure, or through the admixture of "live" superheated steam. The idea is to get the exhaust at the point desired in what is practically a saturated state.

As a general rule, the heat value of "exhaust" steam is taken as equal to 80 per cent. of that of "live" steam at a corresponding pressure, and whilst this forms a safe working basis under ordinary conditions, it can be often taken somewhat greater. The quality of the exhaust from an engine is principally governed by the quality of the entering steam, by the balance or otherwise of the cylinder condensation and re-evaporation, and by the pressure at the end of the stroke.

For estimating the heat value of "exhaust" in terms of live steam at an equivalent pressure, and for other particulars in connection with same, the following formulæ are given:—

$$U_1 = w(L_1 q_1 + S_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

$$U_3 = \frac{U_1 - U_2}{w} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$U_e = q_2(U_3 - S_2) + S_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

or

$$U_e = \left[q_2 \left\{ \frac{U_1 - 2545}{w} - S_2 \right\} \right] + S_2 \quad . \quad . \quad . \quad . \quad (11)$$

and

$$Q = \frac{100 U_e}{L_2 q_3 + S_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

where U_1 = total heat of steam above 32° F. that enters an engine per indicated horsepower-hour,

U_2 = heat converted into work per indicated horsepower-hour $= \dfrac{60 \times 33{,}000}{778} = 2545$ B.Th.U.,

U_3 = heat per lb. of exhaust above 32° F., assuming it dry and saturated,

U_e = heat in 1 lb. of exhaust of any given quality above 32° F.,

w = lbs. of steam per indicated horsepower-hour,

L_1 = latent heat of steam corresponding with the pressure at the admission of cylinder,

L_2 = latent heat corresponding with pressure of exhaust,

S_1 = heat to raise 1 lb. of water from 32° F. to the temperature of the steam entering engine,

S_2 = heat to raise 1 lb. of water from 32° F. to temperature corresponding with pressure of exhaust steam,

q_1 = quality of steam entering engine as expressed by dryness fraction,

q_2 = quality of exhaust steam as expressed by dryness fraction,

q_3 = quality of low-pressure steam of a heating boiler,

Q = percentage heat value of 1 lb. of exhaust steam when compared with live steam at an equivalent pressure.

Example 11.—Determine the value of the exhaust when the conditions are as follows: An engine exhausting into a heating system uses 30 lb. of steam per indicated horsepower-hour at 120 lb. per square inch (gauge pressure), quality 98 per cent., gauge pressure of exhaust 3 lb. per square inch, whilst its quality is 96 per cent.

By formula 8 the total heat of the steam per horsepower-hour is found—

$$U_1 = w(L_1q_1 + S_1)$$

From the steam table in the Appendix the values of L_1 and S_1 for a gauge pressure of 120 lb. per square inch may be taken as 870 and 322 respectively. The dryness fraction is 0·98.

Substituting values,

$$U_1 = 30 \times \{(870 \times 0{\cdot}98) + 322\}$$

when $U_1 =$ **35,238 B.Th.U.**

Heat in 1 lb. of exhaust is obtained by formula 11.

$$U_e = \left[q_2\left\{\frac{U_1 - 2545}{w} - S_2\right\}\right] + S_2$$

From the steam table, the value of S_2 for a gauge pressure of 3 lb. per square inch is, say, 190, whilst that of q_3 is 0·96.

Substituting values—

$$U_e = \left[0{\cdot}96\left\{\frac{35{,}238 - 2545}{30} - 190\right\}\right] + 190$$

$$U_e = (0{\cdot}96 \times 900) + 190$$

when $U_e =$ **1054 B.Th.U.**

Finally, the percentage heat value of the "exhaust" in terms of live steam at the same pressure is obtained by formula 12

$$\text{where } Q = \frac{100\ U_e}{L_2 q_3 + S_2}$$

For 3 lb. gauge pressure the value of L_2 is, say, 964, whilst the quality of the steam will be taken as 98 per cent.—

$$Q = \frac{100 \times 1054}{(964 \times 0{\cdot}98) + 190}$$

when $Q =$ **92·8, or nearly 93 per cent.**

Thus for this case, the value of the "exhaust" when compared with that of live steam is fairly high.

It will be observed that formula 8 is only applicable to problems in which saturated steam is used, but when the steam is superheated this formula may be modified to take the following form:—

$$U_1 = w(H_1 + t_s o) \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

where U_1 = total heat of steam above 32° F. that enters an engine per indicated horsepower-hour,

w = weight of steam in lb. per horsepower-hour,

H_1 = total heat of saturated steam above 32° F.,

t_s = degrees of superheat,

o = specific heat of steam.

The specific heat of steam is not a constant quantity, but it increases as the pressure is raised, and diminishes as the degree of superheat is increased. For approximations, however, its value may be taken as 0·5, and where greater accuracy is necessary, the reader is referred to Marks and Davis, "Steam Tables and Diagrams."

Example 12.—If steam enters the cylinder of an engine at 120 lb. per square inch gauge pressure, and with 100 degrees of superheat, ascertain the relative value of the exhaust when the remaining conditions are as follows: Steam consumption 26 lb. per indicated horsepower-hour, pressure of the exhaust 1 lb. per square inch (gauge), and its quality 98 per cent.

Total heat entering engine per horsepower-hour is found by formula 13.

$$U_1 = w(H_1 + t\,\sigma).$$

For a gauge pressure of 120 lb. per square inch H_1 from the steam table = 1192 B.Th.U.

Substituting values—

$$U_1 = 26 \times (1192 + 100 \times 0{\cdot}5)$$

$$\text{when } U_1 = \mathbf{32{,}292\ B.Th.U.}$$

Heat in 1 lb. of "exhaust." By formula 11—

$$U_e = \left[q_2\left\{\frac{U_1 - 2545}{w} - S_2\right\}\right] + S_2$$

For 98 per cent. the value of $q_2 = 0{\cdot}98$, whilst for 1 lb. gauge pressure $S_2 = 184$ B.Th.U.

Substituting values—

$$U_e = \left[0{\cdot}98 \times \left\{\frac{32{,}292 - 2545}{26} - 184\right\}\right] + 184$$

$$U_e = (0{\cdot}98 \times 960) + 184$$

$$\text{when } U_e = \mathbf{1125\ B.Th.U.}$$

If now the latter value is expressed as a percentage of the heat in "live" steam at 1 lb. gauge pressure, with a dryness fraction of 0·98, then by formula 12—

$$Q = \frac{100\,U_e}{L_2 q_2 + S_2}$$

The value of L_2 for 1 lb. gauge pressure = 968.

$$Q = \frac{100 \times 1125}{(968 \times 0{\cdot}98) + 184}$$

when Q = **99·2 per cent.**

Thus in this case, the heat value of the exhaust is nearly equivalent to that of live steam at the same pressure.

Earlier in this chapter, an example is given to show the economy that may be gained under favourable conditions by the use of exhaust steam. The chief difficulty, however, in a case of this kind is to ascertain beforehand the increased consumpt of steam when back pressure is placed upon an engine. There is no exact mathematical rule for determining it, but where the exhaust from a reciprocating engine is turned into a heating system, the steam consumption may be considered as proportional to the point of cut-off when the initial amount is known for normal working conditions. In other words, where back pressure is added, the mean effective pressure on the piston is assumed as being kept constant by altering the cut-off, whilst the increased consumpt is taken as proportional to this. Under ordinary circumstances, the consumption of steam thus obtained would be greater than the actual one, but the error is on the right side.

The following formulæ and table will aid in determining the increased steam consumpt when back pressure is added to a reciprocating engine, but, as indicated above, the results obtained are only approximate, for correct ones can only be determined by experiment.

$$p_m = \text{PK} - \text{P}_b \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

$$\text{K} = \frac{p_m + p_b}{\text{P}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

$$\text{W}_1 = \frac{w\text{C}_2}{\text{C}_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

where P = absolute pressure of steam entering cylinder of engine in lb. per square inch,

p_m = mean effective pressure in cylinder in lb. per square inch (absolute),

P_b = absolute back pressure when engine is exhausting under normal conditions,

p_b = absolute back pressure in lb. per square inch when exhausting into heating system,

K = cut-off value (see Table V.),

w = steam consumption in lb. per indicated horsepower-hour when engine is operating under normal conditions,

W_1 = steam consumption in lb. per indicated horsepower-hour when engine is exhausting into heating systems,

C_1 = portion of stroke at which steam is cut off when operating under normal conditions,

C_2 = portion of stroke at which steam is cut off when exhausting into heating system.

TABLE V.

Fraction of stroke at which steam is cut off	Value of K.	Fraction of stroke at which steam is cut off	Value of K.
$\frac{1}{6}$	0·465	$\frac{3}{5}$	0·907
$\frac{1}{5}$	0·522	$\frac{5}{8}$	0·919
$\frac{1}{4}$	0·596	$\frac{7}{10}$	0·949
$\frac{1}{3}$	0·699	$\frac{3}{4}$	0·966
$\frac{3}{8}$	0·744	$\frac{4}{5}$	0·978
$\frac{2}{5}$	0·766	$\frac{7}{8}$	0·992
$\frac{1}{2}$	0·846	$\frac{9}{10}$	0·995

Example 13.—An engine when supplied with steam at 90 lb. per square inch (absolute pressure), and when operating against an absolute back pressure of 5 lb. per square inch, absorbs 30 lb. of steam per indicated horsepower-hour, the cut-off being at one-third the stroke. Determine the approximate consumption when the engine exhausts into a heating system against an absolute back pressure of 17 lb. per square inch, and when developing the same power.

Firstly, ascertain roughly the mean effective pressure on the piston by formula 14.

$$p_m = PK - P_b$$

The value of K for $\frac{1}{3}$ cut-off is given in Table V. as 0·699. Substituting value—

$$p_m = (90 \times 0{\cdot}699) - 5$$

when p_m = **57·9 or say 58 lb. per square inch absolute.**

Secondly, obtain value of K when engine is exhausting into heating system by formula 15.

$$K = \frac{p_m + p_b}{P}$$

Take p_m and P as above, whilst p_b is given as 17 lb. per square inch absolute back pressure. Substituting values—

$$K = \frac{58 + 17}{90}$$

when K = 0·833.

In Table V. the nearest value of K agreeing with 0·833 is for $\frac{1}{2}$ stroke. Assume, therefore, the latter as the new point of cut-off.

Thirdly, the increased steam consumption is now derived by formula 16, where

$$W_1 = \frac{w C_2}{C_1}$$

$$W_1 = \frac{30 \times \frac{1}{2}}{\frac{1}{3}}$$

when W_1 = **45 lb. of steam per indicated horsepower-hour.**

Relative Cost of Exhaust Heating.—This may be expressed as follows :—

$$\left.\begin{array}{l}\text{Relative}\\ \text{cost of}\\ \text{exhaust}\\ \text{heating}\end{array}\right\} = \frac{\text{extra cost of fuel consumed to produce a given heating effect when engines exhaust into heating systems}}{\text{cost of fuel to produce equivalent results when “live” steam is used}}$$

Where the “exhaust” is ample, or more than enough to supply the heating surfaces, its percentage cost in terms of “live” steam heating may be obtained by the following formulæ :—

Comparison of "live" and "exhaust" steam heating when the former is generated in low-pressure boilers—

$$x = \frac{100{,}000\,\mathrm{I}(\mathrm{W}_1 - w)f_2}{f_1\mathrm{RU}r} \quad . \quad . \quad . \quad . \quad (17)$$

or

$$x = \frac{100\,\mathrm{I}(\mathrm{W}_1 - w)(1000 + t)f_2}{f_1\mathrm{RU}r} \quad . \quad . \quad (18)$$

Comparison of "live" and "exhaust" steam heating when the former is generated in high-pressure boilers—

$$x = \frac{100{,}000\,\mathrm{I}(\mathrm{W}_1 - w)}{\mathrm{RU}} \quad . \quad . \quad . \quad . \quad . \quad (19)$$

or

$$x = \frac{100\,\mathrm{I}(\mathrm{W}_1 - w)(1000 + t)}{\mathrm{RU}} \quad . \quad . \quad (20)$$

where x = percentage cost of "exhaust" in terms of "live" steam heating,

I = indicated horse-power of engine,

W_1 = lb. of steam consumed per horsepower-hour when engine exhausts into heating system,

w = lb. of steam consumed per horsepower-hour when engine operates under normal conditions.

f_1 = lb. of water evaporated per pound of fuel in high-pressure boiler,

f_2 = lb. of water evaporated per pound of fuel in heating boilers from and at the same temperature as the above,

t = number of degrees water of condensation is cooled below 212° F.,

R = total heating surface in square feet,

U = heat transmitted per square foot of heating surface.

r = relative cost of fuel used in heating boilers to that consumed in high-pressure boilers.

Example 14.—It is proposed to use the exhaust from a non-condensing engine of 50 indicated horse-power. The steam enters the cylinder at 70 lb. per square inch (gauge pressure), the cut-off is made at two-fifths the stroke, steam consumption 40 lb. per horsepower-hour, and the back pressure when exhausting to the atmosphere 3 lb. per square inch. If the heating system contains 1500 square feet of surface, which transmits

250 B.Th.U. per square foot per hour, what is the probable economy when compared with "live" steam heating at 4 lb. pressure per square inch, and when a low-pressure boiler is used?

Assume that the back pressure when exhausting into the heating plant is 5 lb. per square inch, the evaporation per pound of fuel for the power and low-pressure boilers 9 lb. and 8 lb. respectively, whilst the cost of the fuel for a given effect with heating boilers is *one and a quarter* times greater than in the case of the high-pressure boilers.

The increased steam consumption on account of the additional back pressure may be roughly determined as shown, although for such a small difference in the working conditions, the consumpt may be assumed as increased by 8 per cent. when exhausting into the heating plant. If the latter is adopted, W_1 will be 43 lb. per horsepower-hour.

By formula 17

$$x = \frac{100{,}000\,I(W_1 - w)f_2}{f_1 RUr}$$

$$x = \frac{100{,}000 \times 50 \times (43 - 40) \times 8}{9 \times 1500 \times 250 \times 1{\cdot}25}$$

when x = **28 per cent.**

Example 15.—A condensing engine when operating under a vacuum of 24 inches develops 100 I.H.P. and consumes 28 lb. of steam per horsepower-hour. The steam enters the cylinder at an absolute pressure of 90 lb. per square inch, the cut-off being at $\frac{3}{8}$ the stroke. Estimate the probable fuel economy or loss that would result by combining the power and heating plants instead of taking "live" steam for heating direct from the high-pressure boilers; the remaining conditions are as follows: Heating system an atmospheric one, containing 4500 square feet of heating surface; average heat transmitted per square foot of surface, 220 B.Th.U.; back absolute pressure when engine is exhausting into heating system, 19 lb. per square inch; temperature of condensation leaving heating surfaces, 180° F. When "live" steam is used, assume its pressure is reduced to

1 lb. per square inch (gauge pressure) before entering heating system.

Estimate roughly the consumption of the steam when engine exhausts into heating plant by formulæ 14, 15, and 16.

$$(a) \qquad p_m = \text{PK} - \text{P}.$$

24 inches of vacuum equal approximately 3 lb. per square inch. absolute pressure, and the value of K for [illegible] cut-off = 0·744.

Then $p_m = (90 \times 0{\cdot}744) - 3$.

when $p_m =$ **63·96 as the mean effective pressure.**

$$(b) \qquad \text{K} = \frac{p_m + p_b}{\text{P}}$$

Here p_b is 19 lb. per square inch, and

$$\text{K} = \frac{63{\cdot}96 + 19}{90}$$

when $\text{K} =$ **0·922**.

Glancing at Table V., the nearest value of K to 0·922 is that for [illegible] **cut-off**, which will be taken as the new point for estimating the steam.

$$(c) \qquad \text{W}_1 = \frac{w\text{C}_2}{\text{C}_1}$$

$$\text{W}_1 = \frac{28 \times [illegible]}{[illegible]}$$

when $\text{W}_1 =$ **47 lb. per horsepower-hour**.

The loss or gain can now be ascertained by formula 20, where

$$r = \frac{100l(\text{W}_1 - w)(1000 + t)}{\text{RU}}$$

For the example given, the steam would be superheated in flowing through the reducing valve. The effect of this, however, may be omitted so far as the calculation is concerned. The condensation would be cooled through 212 − 180 = 32° F.

Substituting values

$$x = \frac{100 \times 100 \times (47 - 28) \times (1000 + 32)}{4500 \times 220}$$

when $x =$ **198 per cent.**

Here the cost of "exhaust" heating is shown to work out at 98 per cent. in excess of "live" steam heating. The engine, therefore, in the example, is much too large to be converted into a non-condensing one for the amount of heating surface and the rate of heat transmission given.

CHAPTER XII

EXHAUST STEAM HEATING (*continued*)

De-oiling of Exhaust Steam.—When the water of condensation is returned to a boiler, some appliance is desirable by which the lubricating oil and other impurities may be extracted from the "exhaust" before being transmitted to a heating system, for should the heating surfaces get coated with a film of grease, their efficiency for transmitting heat would be appreciably impaired. Various appliances are in use for the de-oiling of exhaust steam, but when used in connection with heating plants, no unnecessary back pressure should be put upon the engines.

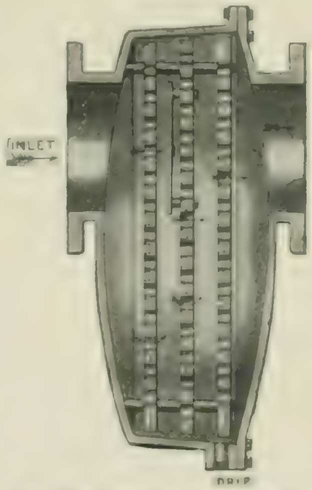


FIG. 98.—Kieley and Mueller's grease separator.

In Fig. 98, a common form of grease extractor or de-oiler is shown, in which a series of ribs, plates, or other construction is introduced against which the steam will strike, the greasy matter either accumulating upon these surfaces, or trickling to the bottom of the appliance. Through the drip pipe, the oily matter is passed to waste, its discharge being regulated either by a siphon or a steam trap, according to the pressure of the steam.

Fig. 99 gives a muffler and de-oiler, an appliance which is advantageous on account of the large area it provides for the interception of oil, whilst the space available, in a great measure

tends to compensate for the irregular discharge of the exhaust steam from an engine. In Fig. 99, chain baffles are used for the de-oiling of the steam, but other forms of fixed baffles are also used. Mufflers and de-oilers may be obtained in either vertical or horizontal forms, and they may also be combined with feed water heaters and pump governors and receivers.

Oil separators with fixed parts are more or less liable to

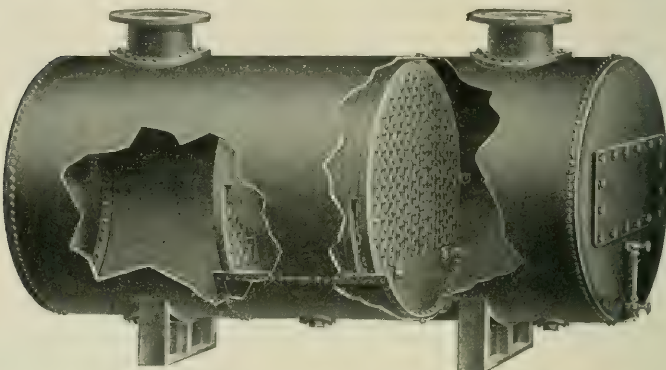


FIG. 99.—The Utility exhaust muffler and oil separator.

permit some of the greasy particles to be passed through them with the steam, and for this reason, care should be taken to choose the best types and those that can be readily cleansed.

In order to avoid oil being carried through separators, movable types have been designed, these being either operated with the exhaust steam or driven by some external power. Fig. 100 gives a self-acting separator in which the cylinder *c* is made to revolve by the steam flowing from the fixed chamber B, and through the upper row of ports indicated by *a*. The oil which is caught by the revolving baffles, is driven by centrifugal force to and down the sloping walls of the outer cylinder C, from which it is discharged into the chamber below through the apertures shown at E. The steam flows to the upper part of the appliance. and through the apertures D to the outlet. The two rows of ports in chamber B are differently formed, the upper ones being diagonally cut, whilst the lower slots are straight. By this arrangement, the speed of rotation can be regulated by the hand-wheel F, which alters the proportion of

the diagonal and straight slotted ports of chamber B in contact with those of cylinder c.

Revolving separators will necessarily offer more resistance to the flow of steam than the usual fixed form, but a well-

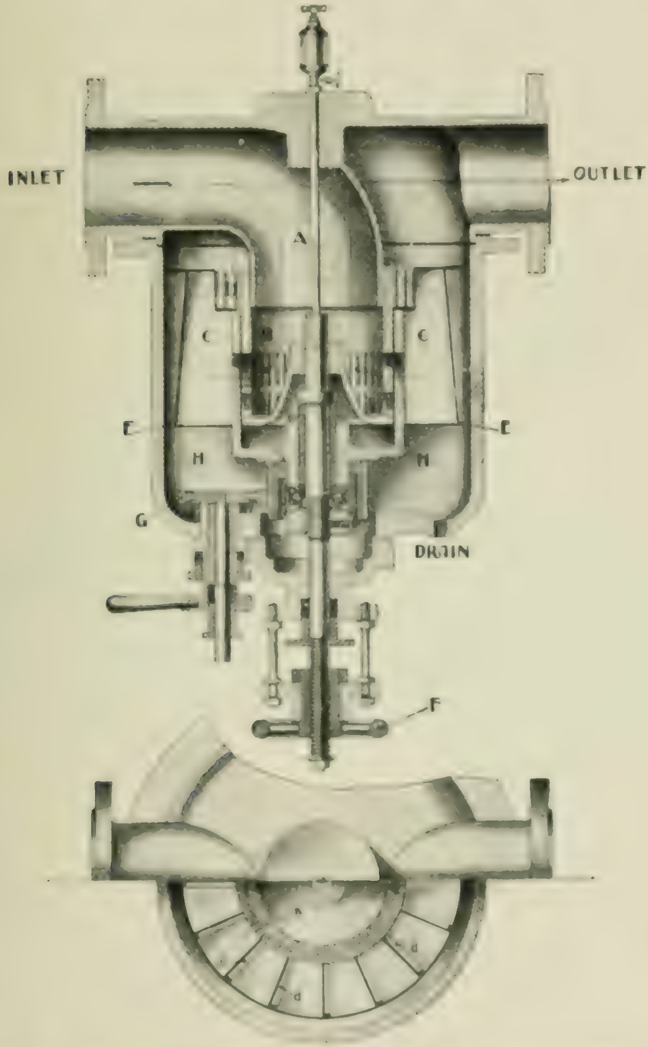


FIG. 100.—Centrifugal grease extractor. By Messrs. Lassen and Hjort.

designed rotary type should not offer a back pressure exceeding ½ lb. per square inch.

Back-Pressure Valves.—To a great extent, the efficiency of exhaust heating depends upon the back-pressure valve that is used, for, whilst it should offer enough resistance to drive the steam through the heating plant, yet the back pressure should not be raised beyond a predetermined amount. The action of these valves is of special importance where the exhaust for certain periods must be supplemented by " live " steam, for should they be unreliable in action, much steam may escape to waste.

There are three principal forms of back pressure valves, viz. the "single," "double or balanced," and the "multiple" types. The best to use depends very much upon the size required, and upon the conditions to be satisfied. Generally speaking, the single-seated valves of good design are the best for diameters up to about 6 inches, whilst the "balanced" and multiple valves are more suitable for the larger sizes, Fig. 101 shows a single-seated back-pressure valve. Here it

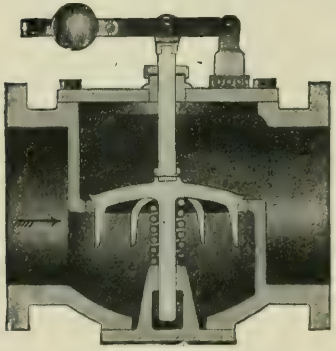


Fig. 101.—Kieley and Mueller's back-pressure valve.

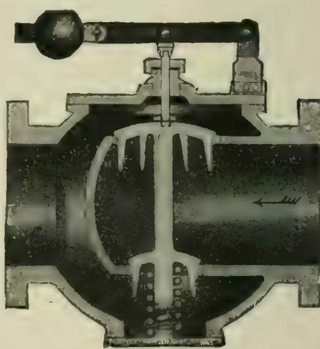


Fig. 102.—Kieley and Mueller's back-pressure valve.

will be seen that the load to be concentrated upon the disc must be a little in excess of the total pressure it is intended to resist. For example, if the valve surface exposed to the steam has an area of 38 square inches and a maximum back pressure of 3 lb. per square inch is allowed, the total load to be concentrated upon the disc must equal $38 \times 3 = 114$ lb. It is important that the disc should not be dashed against its seating

by being too suddenly closed, but this can be guarded against by the provision of dash-pots and springs, or by other means.

In Fig. 102, a double-seated valve is shown in which a portion of the pressure acting upon the upper disc is neutralized by the pressure on the lower disc. With this form of valve, smaller regulating weights can be used, and the lift of the discs reduced. On the other hand, double valves are more difficult to keep in order, as they are liable to stick.

A multiple form of back-pressure valve is given in Fig. 103, a series of small single-acting discs being arranged on one large

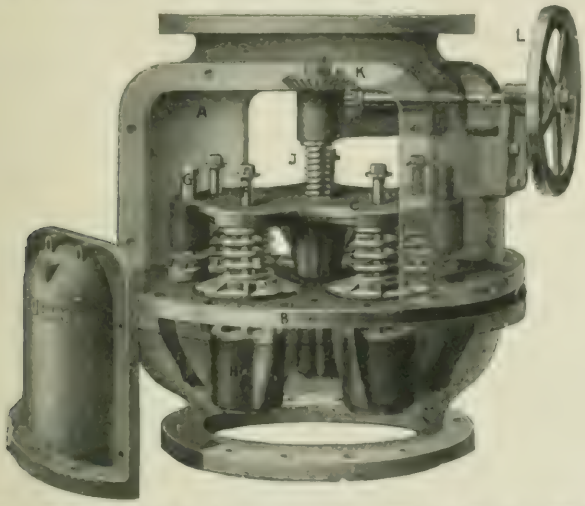


FIG. 103.—Multiple back-pressure valve. By the Harrison Safety Boiler Works.

decking. The merits of well-constructed multiple valves consist in the small lift essential for their discharge, and the lessened risks of their being damaged. Assume, for example, a single-seated valve of 12 inches diameter being used where the back pressure carried is 5 lb. per square inch. Here the total load upon the disc must be not less than $12^2 \times 0{\cdot}7854 \times 5 = 565{\cdot}48$ lb., whilst the load upon each lineal inch of its facing would be $\frac{565{\cdot}48}{12 \times 3{\cdot}1416}$, or nearly 15 lb. If now a multiple valve were used that has six discs of 5 inches diameter, the concentrated

load upon each would be $5^2 \times 0{\cdot}7854 \times 5 = 98{\cdot}17$ lb., whilst that on each lineal inch of the facing would be $\frac{98{\cdot}17}{5 \times 3{\cdot}1416}$, or rather more than 6 lb. The discs of back-pressure valves, in order to give a full discharge, require to be raised through a height of one-fourth their diameters when the coefficient of discharge is taken as unity. In reality the coefficient is less than unity, and varies with the different valves, so that in practice the lift is less than stated. For comparative purposes, however, the discharge coefficient may be considered as unity, so for discs of 12 inches and 5 inches diameter, their lifts would be 3 inches and $1\frac{1}{4}$ inch respectively. From the above, it will be observed that in the case of the smaller valves not only is the force on each linear inch of seating reduced, but the possible destructive effect is also diminished by the shorter drop. At the same time, the smaller discs have greater freedom to respond.

The appliance shown in Fig. 103 is of ingenious design, the valves being spring controlled and provided with dash-pots which have water cushions. To permit of more or of less loading of these valves, a pressure plate is used to which the upper part of the springs is attached. Regulation is effected by the hand-wheel at the side of the fittings, but from the construction of the pressure plate, it will be seen that the springs can only be compressed to a limited extent. The discs may also be raised above their seatings when it is desired to open the exhaust piping to the atmosphere.

Feed-Water Heaters.—The advantages of heating feed water are generally recognized, so that where economizers are not installed the heating of this water may be combined with the "exhaust" heating equipment.

Feed-water heaters are of two principal types, viz. closed and open ones. In the first, the temperature of the water is raised by contact with heated surfaces, whilst in the second, the steam and water are brought together.

Fig. 104 gives a closed or surface heater, the "exhaust" being discharged into the casing through either the upper or lower connection. It is good practice to make the water and steam connections so that these media flow in opposite ways.

Surface heaters vary widely in constructional details, the most efficient forms being those that condense the greatest weight of steam per unit area of surface, and offer the least back pressure.

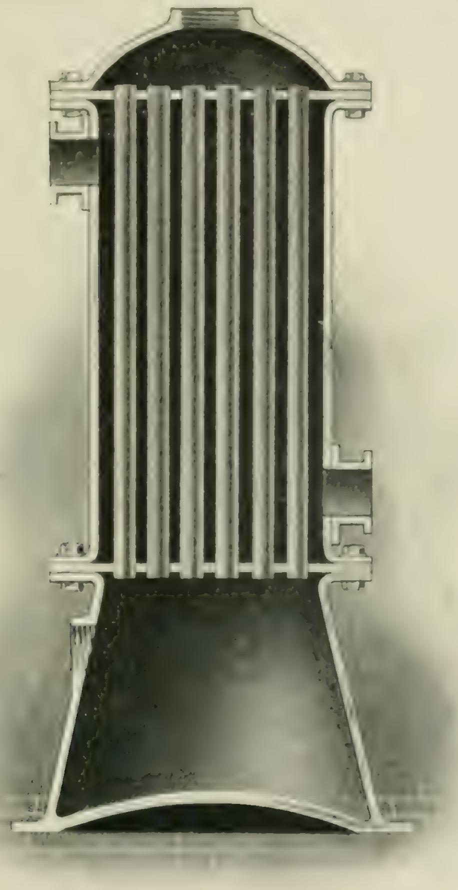


FIG. 104.—Closed feed-water heater.
By Messrs. Lumley, Son, & Wood, Ltd.

"Exhaust" steam heaters should admit of being readily cleansed, and this is essential where the oily matter is largely intercepted by them. Where straight tubes are used, as in Fig. 104, due

provision should be made for their expansion, otherwise leakages will arise. These heaters may be located in any convenient place in the "exhaust" line, the condensation being simply pumped through them on its passage to the boiler. The temperature to which the condensation can be raised may be made to approach within a few degrees that of the exhaust steam.

Open Heaters.—In Fig. 105, an open heater is shown, with

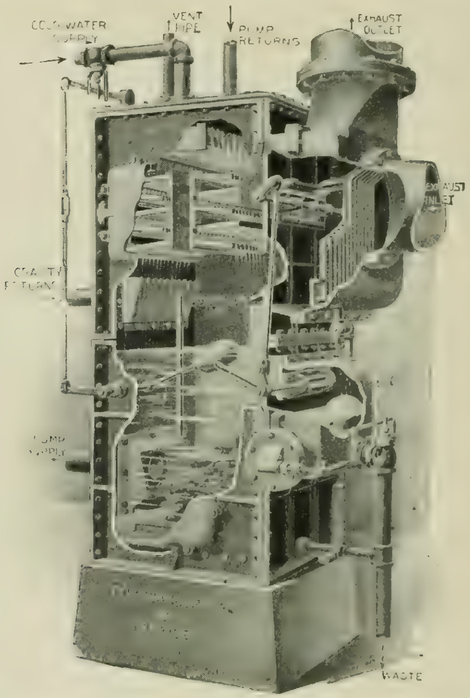


FIG. 105.—Open feed heater. By the Harrison Safety Boiler Works.

which is combined a grease separator, receiver, and filter. By means of a rotary valve, the whole of the "exhaust" steam can be diverted through the outlet, when it is desired to put the heater out of use. The cold supply "make up" water is controlled by the float on the left of the figure, and by means of trays or spreaders, this water is broken up so as to come into intimate contact with the steam. From the separator, the

greasy matter drips to the trap beneath, whilst into the same trap, the overflow water from the receiver is also discharged. The gravity returns are shown to enter a little below the overflow line, provision being made for sealing these to prevent the entry of steam from the heater. At the base of the appliance, a coke filter is placed, by which any foreign matter may be extracted from the water in its passage to the feed pump. When the cut-out valve is open, the more direct passage for the "exhaust" is into the body of the heater, the rate of condensation being roughly proportional to the volume of feed water to

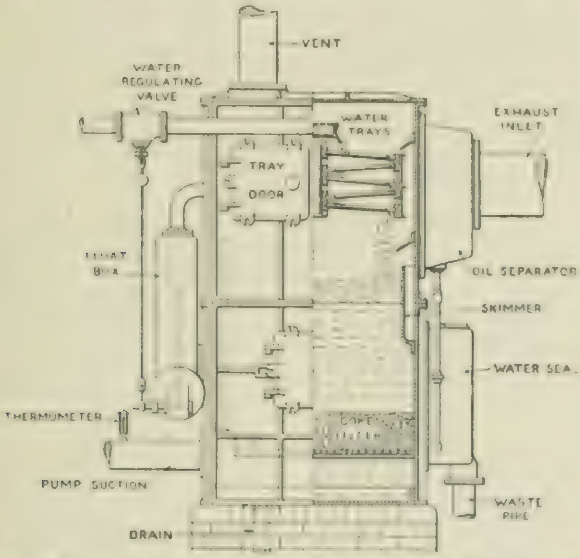


Fig. 106.—Open feed-water heater. By Erith Engineering Co.

be added. Where the whole of the condensation from a heating plant is returned, only a small weight of the "exhaust" steam would be utilized at the heater. On the other hand, with a non-condensing engine and the heating apparatus out of use, the volume of fresh water to be added must necessarily be equal to the rate of evaporation. In the latter case, less than one-fifth of the heat of the "exhaust" is used in raising the whole of the feed water to over 200° F.

Another open heater is shown in Fig. 106, the general

construction being similar to that of the previous figure, but differing in a few of its details. No connections are indicated in Fig. 106 for the return mains, but these can be readily provided.

With atmospheric systems of "exhaust" heating, and where the condensation is very appreciably cooled before arriving at the heater, it would be advantageous for the returned water to enter at the top of the heater so as to be heated by mingling with the steam. In some cases, it may be possible to locate the heaters so as to give a gravity flow, but in others, the condensation would require to be pumped, especially where the heaters were subjected to a little internal pressure.

As the term implies, the heaters shown in Figs. 105 and 106 are open to the air, vents being provided at their highest points.

Condensation Return and Boiler Feed Pumps.—The pumps for handling the feed water to boilers may be classified as "steam driven" and "electrically driven" types. Steam pumps of the piston class may be further divided into "single" and "duplex" forms, and these may be either horizontally or vertically arranged.

Steam Pumps.—For returning the condensation from heating apparatus, the piston type is in general use, owing to the simple manner in which the speed of the pumps can be automatically effected, whilst the "exhaust" from these can be passed to the heating system. In the usual form of duplex pump, two pumps are placed side by side, and are so arranged that the piston of the one operates the slide valve of the other. After a piston has finished its stroke, it remains at that point until its slide valve is opened by the adjoining piston, when steam is admitted for another stroke. The chief merits of duplex pumps consist in their simplicity, the small amount of attention required, and their comparatively low cost; their chief drawback is due to the fact that they are rather extravagant in the use of steam, but this feature is of no moment when the "exhaust" can be utilized in the heating system.

A pump of a higher grade, and one suitable for working against a heavy pressure is shown in Fig. 107. The illustration gives a pair of pumps, and both may be operated at the same time, or either may be used as a "stand by." Although

these operate as "single" double-acting pumps, there is no dead centre, so that they will start or stop at any part of the stroke.

Electrically Driven Pumps.—It occasionally happens, when

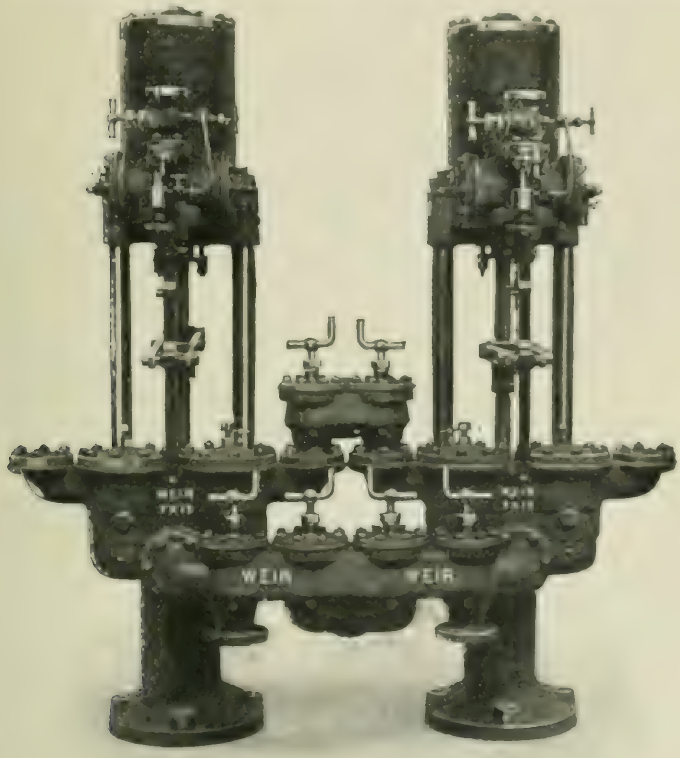


FIG. 107. Boiler feed pumps. By Messrs. G. & J. Weir, Ltd.

handling the condensation from heating apparatus, that an electrically driven pump is the more suitable to use. Where the delivery is against a high pressure, and a piston pump is adopted, the "triplex" form is a suitable one, as it practically delivers a constant stream, and in turn diminishes the strain and shock incidental to crank-action pumps of fewer cylinders.

If, however, it is only essential for a pump to work against a low head, a duplex type will answer very well.

A "triplex" pump with electric drive is indicated in Fig. 108, and in conjunction with it, is shown a receiver. By means

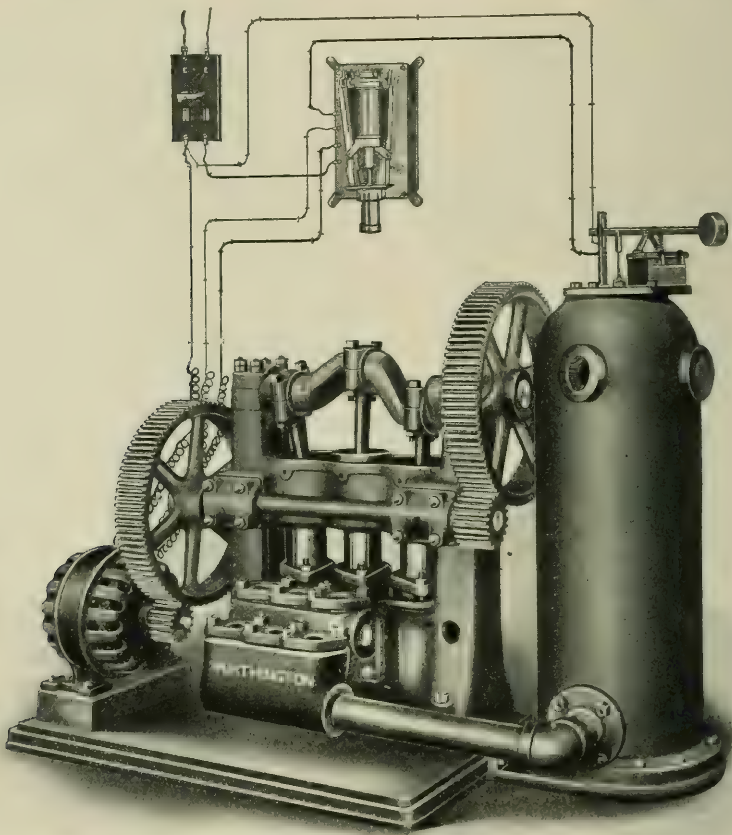


FIG. 108.—Triplex boiler feed pump. By the Worthington Pump Co.

of a float in the receiver, motion is imparted to the lever which opens or closes the electric circuit.

Directly-driven centrifugal pumps are also suitable for returning the condensation and for feeding boilers, and these are

specially serviceable where a continuous and constant discharge is required. To operate against a high boiler pressure, the "Multistage" centrifugal type is used. This consists of two or more impellers working in separate chambers, but keyed to one common shaft. Fig. 109 gives a pump of this form that is

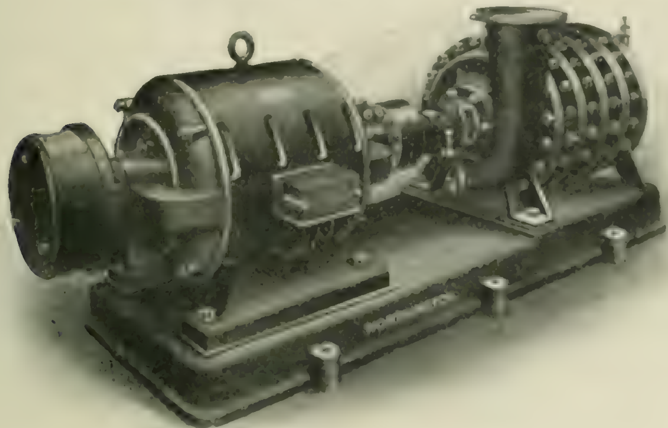


FIG. 109.—Centrifugal boiler feed pump. By the Worthington Pump Co.

connected directly to a three-phase motor. Centrifugal pumps are fairly flexible in operation, for the volume of water discharged may be readily adjusted by means of throttle valves in their delivery pipes. Unlike piston pumps, the throttling of their outlets does not increase the load, but on the contrary diminishes it.

Low-Pressure "Exhaust" Systems.—For "exhaust" steam heating, any ordinary piping arrangement may be adopted. In Fig. 110, a general "lay-out" is shown where provision is made for adding "live" steam when it is required. The piping to the heating surfaces is on the "overhead" or "down feed" principle, and for exhaust heating it is a good system to adopt. From the engine, the exhaust steam flows through the pipe E and into the grease extractor S, whilst just beyond the latter appliance, a valve is placed to divert the steam through the feed-water heater. Above the highest distributing main, the back-pressure valve is located, this being loaded to drive the

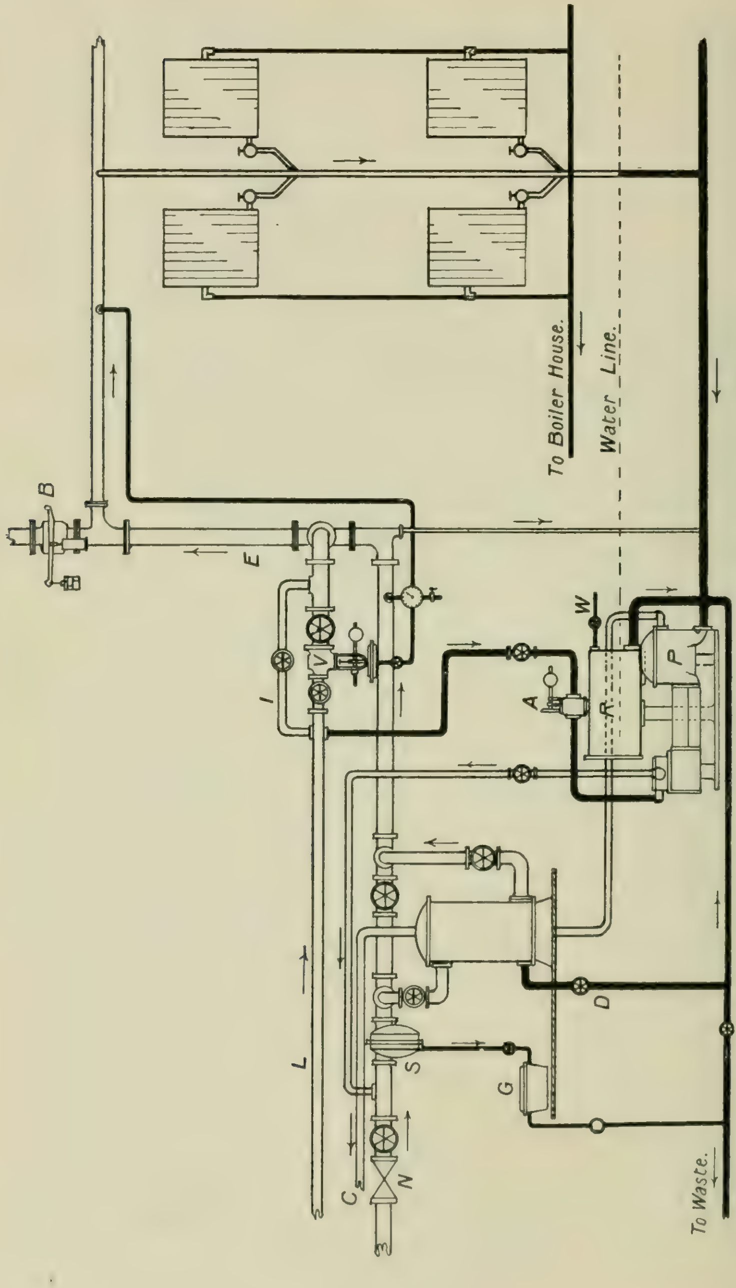


FIG. 110.—Exhaust steam heating system.

B = back pressure valve.
E = exhaust pipe.
I = by-pass.
L = live-steam pipe.
C = pipe discharging condensation to boiler.
N = check valve.
S = grease separator.
V = pressure reducing valve.
G = grease trap.
D = condensation outlet pipe.
A = automatic steam valve.
R = receiver.
W = water-supply pipe.
P = pump for handling condensation.

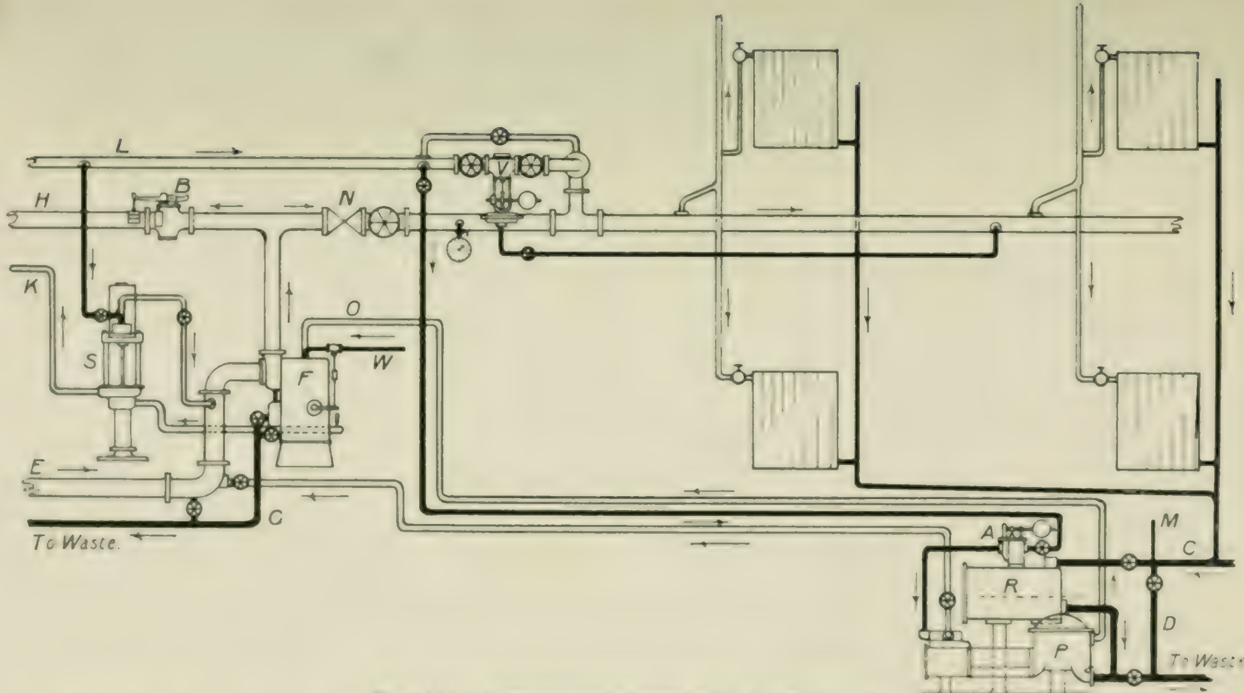


Fig. 111.—Atmospheric exhaust steam heating plant.

L = live steam pipe.
V = pressure reducing valve.
H = exhaust outlet pipe.
B = back pressure valve.
N = check valve.
K = delivery pipe to boiler.
S = boiler feed pump.
F = feed water heater.
W = water supply pipe.
O = condensation pipe from pump and receiver.
E = exhaust pipe.
G = waste pipe.
A = automatic steam valve.
R = receiver.
P = pump for heating system.
M = air pipe for joining with nearest flue.
C = return main.
D = by-pass for condensation.

steam into the heating main. Through the pipe L, live steam is admitted to the heating installation at a reduced pressure of, say, 2 lb. per square inch, while the non-return valve N prevents its escaping in the direction of the engine. A by-pass is provided in the live steam main with the necessary cut-out valves, these being useful when the reducing valve is under repair. From the receiver R, the condensation is returned through the feed heater to the boiler, or to any other point desired, whilst the "exhaust" from the pump is discharged into the main exhaust pipes.

With an ordinary low-pressure system, it is usually essential to increase the back pressure when bringing the heating plant into use, this being reduced to the normal amount when the air has been dislodged from the pipes and heating surfaces. To prevent unpleasant odours being emitted from "exhaust" plants, the automatic air valves should be joined with an "air-line" which discharges at a point where no nuisance would be caused.

Atmospheric Systems of Exhaust Heating.—A general "lay-out" for an apparatus of this class is given in Fig. 111. Let it be assumed that non-condensing engines are in use, that about 60 per cent. of the "exhaust" steam is used in the heating system, and that the receiver R is located at a very low point in order for the condensation to gravitate to it.

Owing to the freedom with which air can be displaced from an atmospheric plant, only a small back pressure is required, and for large systems, the load on the back-pressure valve B need not exceed 1 lb. per square inch. The exhaust pipe E is indicated as discharging into an open heater, the condensation gravitating to the receiver R whilst the air from the latter is discharged into the nearest flue. In the feed-water heater, the "make-up" water is added by means of a float-controlled valve, and from this apparatus, it is delivered to the boiler by the vertical steam pump S.

When the principal engines are not running, "live" steam is supplied through the reducing valve V, the pressure being diminished to say ½ lb. per square inch.

CHAPTER XIII

VACUUM AND VACUO-VAPOUR SYSTEMS OF STEAM HEATING

ALTHOUGH the systems falling under this heading are very numerous, for the purpose of classification they may be divided into two principal groups, viz. the "semi-positive" and the "positive." The "semi-positive" systems are those in which the vacuum created is principally due to the condensation of steam in the heating surfaces themselves, whilst in the case of "positive" systems external agencies are employed, such as injectors, condensers, and pumps.

The "positive" group of vacuum plants may be subdivided into two classes: (*a*) those in which the air and water of condensation from the heating surfaces are conveyed by separate channels; and (*b*) those in which the air and condensation are discharged through one and the same pipe. Falling within the range of the latter subdivision, there are various arrangements, but they chiefly differ in details of construction, and in the manner of producing the vacuum.

Merits and Limitations.—The advantages generally claimed for vacuum systems are—

(1) The effectual removal of air from the radiator or other heating surface, thereby increasing the efficiency of the latter for the transmission of heat.

(2) The efficient drainage of the water of condensation.

(3) Flexibility with regard to temperature regulation and the arrangement of the piping.

(4) Comparatively low working cost.

(5) Freedom from clicking and hammering sounds.

Under favourable conditions, and in well-designed plants, these conditions can be fulfilled, but on the whole, vacuum

systems show to the best advantage when of a large size, where "exhaust" steam is available, and where some difficulty arises in handling the condensation, as *e.g.* when the main return pipe is trapped, or where located at a higher level than some of the heating surfaces.

All so-called vacuum systems do not possess the merits described above, and many people have erroneous ideas as to what may be accomplished by them.

Whether air is effectually removed or not from a vacuum system will greatly depend upon the method adopted, and upon the efficiency of the appliance used. In the "semi-positive" systems, some air will be nearly always retained within the heating surfaces, unless special provision is made for dislodging it when the steam is first turned on.

To admit of effective drainage in "semi-positive" installations, the return pipes require a good pitch. In other words, the piping should conform to the requirements of ordinary gravity systems.

The regulation of temperature by the manipulation of a radiator valve can only be effected in vacuum systems where the condensation and air flow through the same pipe. In all "positive" plants, however, the temperature of the heating surfaces may be varied to a more or less extent by modifying the pressure of the steam supply, or by increasing or decreasing the degree of vacuum.

The degree of vacuum that should be maintained depends upon the plant as a whole, *e.g.* its size, and the free gravitation or otherwise of the condensation from the radiators to the exhauster. As a general rule, however, the degree of vacuum at the most distant point should not be less than from one to two inches of mercury, the higher value being desirable as a minimum where exhaust steam is used.

Limitations.—The failings that arise in vacuum systems are as follows:—

(*a*) Difficulty in maintaining the requisite degree of vacuum.

(*b*) The periodical flooding of return pipes and its attendant noise.

(*c*) The attention necessary to keep the appliances in proper order.

As a general rule, the necessary vacuum can be obtained where pumps are used, but if a system is faulty, the cost of maintaining it may be out of proportion to the benefit gained. In systems where no "positive" means are utilized, the vacuum is quickly destroyed either through leaky supply or air valves. As a matter of fact, the infiltration of air from this source is largely responsible for the loss in efficiency of vacuum plants in general.

The flooding of pipes may be due to the vacuum in the heating surfaces exceeding that produced by the exhauster, or, in the case of certain systems, it may result from the vertical distance between the boiler water-line and the horizontal return main being too small. A defective check valve may also be the cause, and where a "one-pipe" system is installed, the condensation may be held up in a radiator through the branch supply being of too small a bore.

To maintain a vacuum system in good order, it is necessary that the various appliances be periodically overhauled. If the fittings are of simple construction, this involves but little time, whilst it causes them to last longer.

Vacuo-Vapour Systems.—This term applies to those in which the pressure of the steam supply is similar to that of the atmosphere, or a little below it. In the earlier vacuum systems, the reduced pressure was chiefly confined to the heating surfaces and to the return line, whilst any ordinary pressure was carried by the supply mains. Thus in vacuo-vapour systems, a partial vacuum is often created throughout the whole of an installation.

Fig. 112 shows a simple system that operates chiefly through the main air line acting as a condenser, a mercury trap being used at T for sealing the end of the pipe and for releasing air. The system operates as follows: At the outset, the steam pressure is raised to a few pounds per square inch, and is maintained at that for a short time to expel the air from the plant. Afterwards, the steam supply is allowed to fall to a pressure of 1 lb. per square inch or less, whilst its circulation to the radiators is facilitated to some extent by a partial vacuum forming in the air line. The success of this apparatus depends principally upon the use of reliable air valves, and by reducing

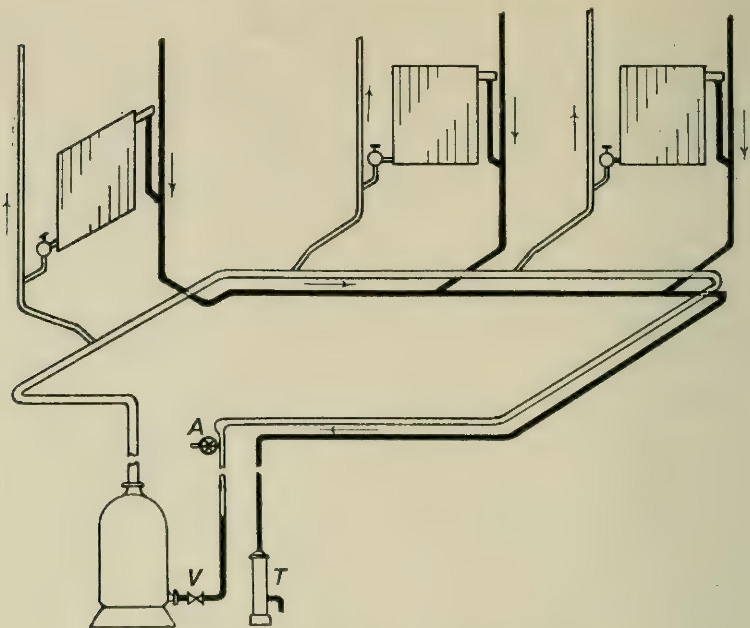


FIG. 112.—Simple form of vacuum system.

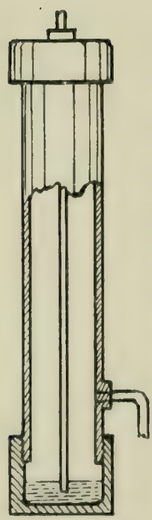


FIG. 113.—Mercury sealing appliance.

to a minimum all points where the infiltration of air may occur.

The construction of the mercury trap is shown in Fig. 113; this consists of an outer and an inner tube, the lower end of the latter dipping into a small reservoir of mercury. Although the mercury seal offers but little resistance to the expulsion of air, its entry at this point is prevented by the mercury rising in the inner tube when a partial vacuum is created.

"Positive" Vacuum Installations with Independent Air Lines

"**Paul System.**"—An installation of the "positive" class is illustrated in Fig. 114, where exhaust steam is the principal heating medium, but provision may also be made for the utilization of "live" steam. For creating the vacuum

at the end of the air line, a steam injector I is shown, whilst the exhausted air is indicated as being discharged into the main steam exhaust pipe E. The piping supplying the steam is on the "overhead" or "down feed" principle, and the lower ends are sealed by joining with the "wet" return. In the system shown, the water of condensation can gravitate to the pump, and only a small back pressure is necessary to aid in the

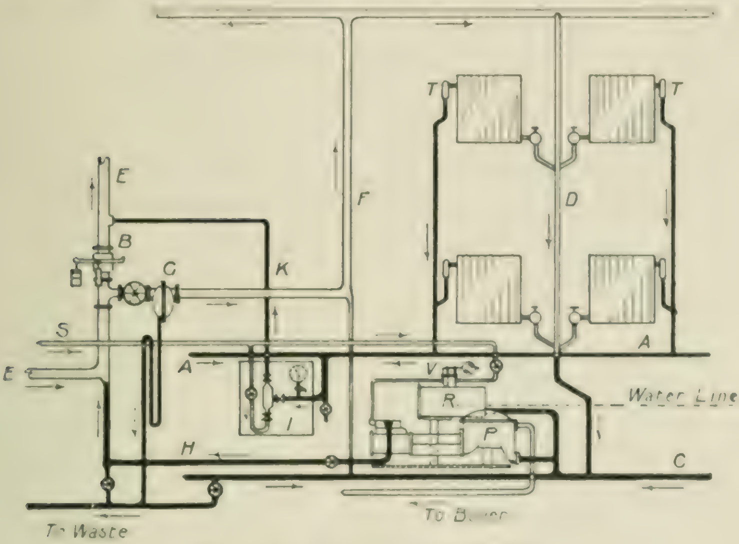


FIG. 114.—"Paul" vacuum system.

E = exhaust pipe.
B = back-pressure valve.
G = grease separator.
S = steam supply to pump.
K = exhaust from injector.
F = steam supply to heating system.
T = automatic air valves.
D = drop pipes.
A = air line.
V = automatic steam valve.
R = receiver.
P = pump.
H = pump exhaust.
C = condensation return pump.
I = injector.

circulation of the steam supply. Before steam is turned into the system, the air is first exhausted by bringing the injector into use. The air valves used are of the expanding type and are closed when in contact with steam. After the apparatus is brought into use, the vacuum has no influence beyond the air valves, that is unless the steam supply is curtailed so as to make it fall short of the maximum condensing capacity of the heating

surfaces. Under the latter conditions, the partial vacuum may extend into and beyond the radiators.

"**Spark's System.**"—Fig. 115 shows another installation in which an independent air line is employed, and, like the previous one, the piping should be so arranged that the con-

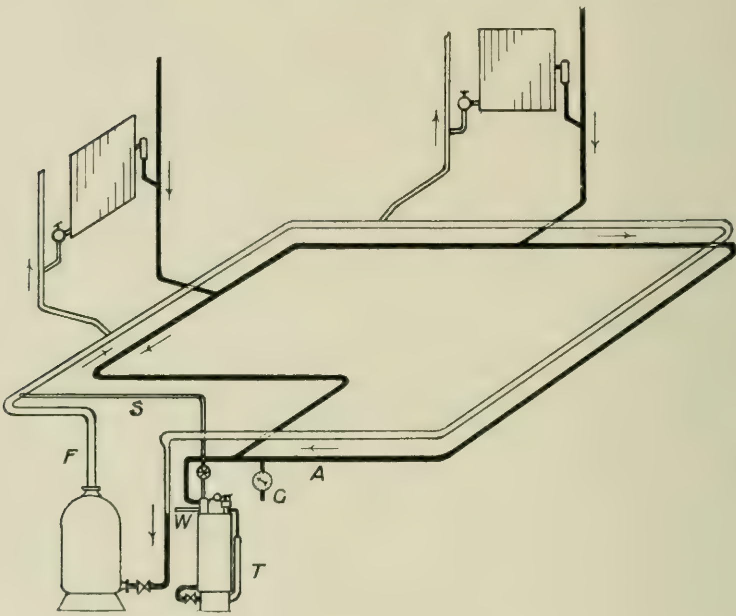


FIG. 115.—"Spark's" vacuum system.

F = steam main.
S = steam supply to exhauster.
W = water supply to exhauster.
A = air line.
G = vacuum gauge.
T = thermostat.

densation can gravitate to the boiler or receiver. The main steam and air lines are shown as they would skirt a basement where a "one pipe" "up feed" system is chosen. The exhauster takes the form of a cylindrical tank in which a partial vacuum is created by the admission and condensation of steam. The action of this exhauster and the mechanism in connection with it will be better understood from Fig. 116. Near to the base of the appliance is located a thermostat, which supplies the power for operating the steam and water valves. During the interval

the steam is entering the tank its pressure will expel any contained air or condensation present, but upon coming in contact

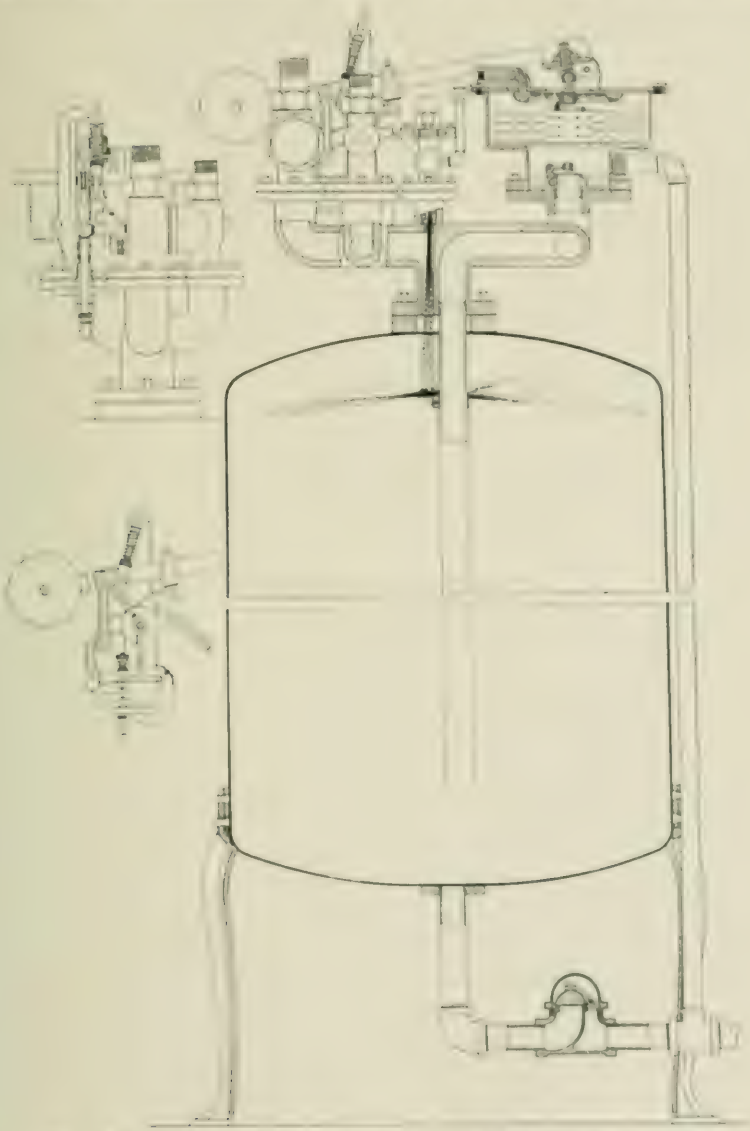


FIG. 116.—Section of "Spark" exhauster. By Automatic Vacuum Co.

with the thermostat, its contained fluid is vaporized, and this transmits pressure to the bellows at the top of the exhauster. At this stage, the steam valve is closed whilst water is admitted to the tank in the form of a spray. A partial vacuum is thus created, and the supply of water cut off. To regulate the number of pulsations in a given time, a locking attachment is provided which prevents the re-opening of the steam valve until the vacuum has fallen to a given degree.

The air valve used in the "Spark-Eddin" system contains a small open aperture, so that the vacuum extends to within the heating surfaces. The use of the condensing water accelerates the action of the exhauster, and permits a partial vacuum to be maintained where a certain amount of leakage occurs. Leakage, however, should be avoided as far as possible, for not only does it interfere with the proper distribution of the steam, but it adds to the cost of operation.

From the foregoing illustrations, it will be observed that any ordinary low-pressure system with an air line may be converted into a vacuum one, by simply attaching some form of exhauster.

"Positive" Systems in which the Water of Condensation and Air flow together

"**Moline**" **System**.—In Fig. 117, a plant is given where the vacuum is created by an injector in conjunction with a condensing radiator. By tracing the pipes, it will be seen that both the steam and return pipes join with the injector, whilst the outlet of the condensing radiator is connected with the return pipe below the water line. The only outlet for the escape of air is indicated at T, and at this point, a combined air and vacuum valve is introduced. For the permanent control of the steam supply, sleeves with regulated apertures are fitted into the radiator valves, whilst a similar form of control is adopted for the radiator returns. To make the injector effective in this system, the steam at that point should be maintained at a fair pressure. This will necessarily vary with the size of the plant, but as a rule it need not exceed 2 lb. per square inch, and 1 lb. will suffice in the smaller installations. To

obviate any trouble through the water of condensation flooding the returns, the pressure due to the vertical distance D should exceed the differential pressure to be maintained in the system. The condensing radiator C should be placed near to the end of the circuit, whilst the vertical distance E between the air valve

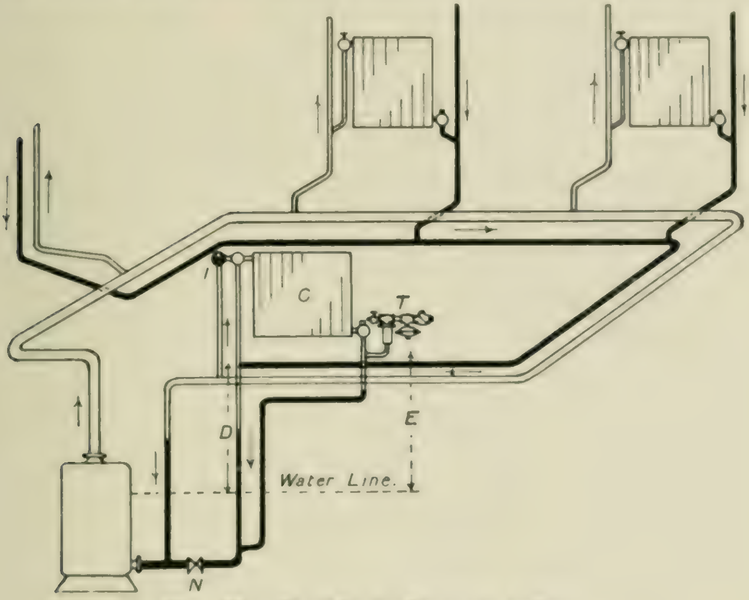


FIG. 117.—The "Moline" vacuum system.

C = condensing radiator. N = check valve.
T = air trap and vacuum valve.

and water-line should be not less than that at D. All the pipes should be pitched so that the condensation can gravitate to the boiler. From the foregoing, it will be observed that although the steam supply may be under two or more pounds per square inch, the pressure within the radiators may be less than that, on account of the throttled supply, and the effect of the injector on the return pipe.

"**Dunham**" **System**.—Another system of the same class is indicated in Fig. 118, its principal features being the means which are adopted for admitting steam to the exhauster, and the special appliances that are used with it. The automatic

air and vacuum valve may be joined with the pipe as shown, or it may be attached directly to the top of the condenser, whilst to provide the necessary outlet area in large plants, two or more combination valves may be grouped at one or more points. An equalizing pipe is shown joined between the boiler and the condensing tank, and the distance between

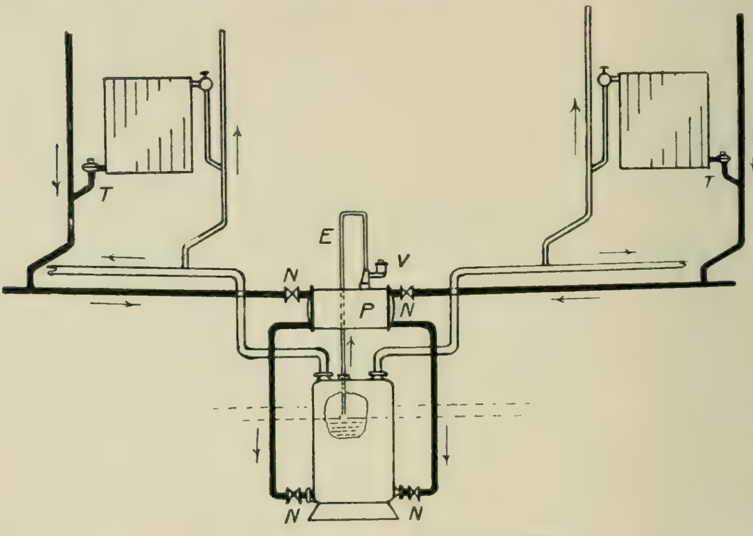


Fig. 118.—" Dunham " vacuum system.

T = radiator trap.
E = equalizing pipe.
V = air and vacuum valve.
P = exhaustor.
N = check valves.

the two water-lines represents the volume to be evaporated before the steam can enter the tank. Assuming the level of the water in the boiler to be at the upper line, the action is as follows: As the evaporation of the water proceeds, the condensation returns and accumulates in the tank until the lower water-line is reached. At this period, steam gains admittance to the tank and equalizes its internal pressure with that of the boiler, when, in virtue of the position of the tank, the condensation re-enters the boiler, raising the water-line, and in turn cutting off the steam to the tank. As the steam condenses in the tank, a partial vacuum is formed which exerts a pull on the return line. By carefully regulating the distance between

the water-lines, the period of action may be adjusted to occur as desired. As a provision against the water returning from the boiler to the tank, the equalizing pipe E is carried at least 6 feet above the water-line before joining it with the tank.

Webster System.—This is shown in Fig. 119, and it is

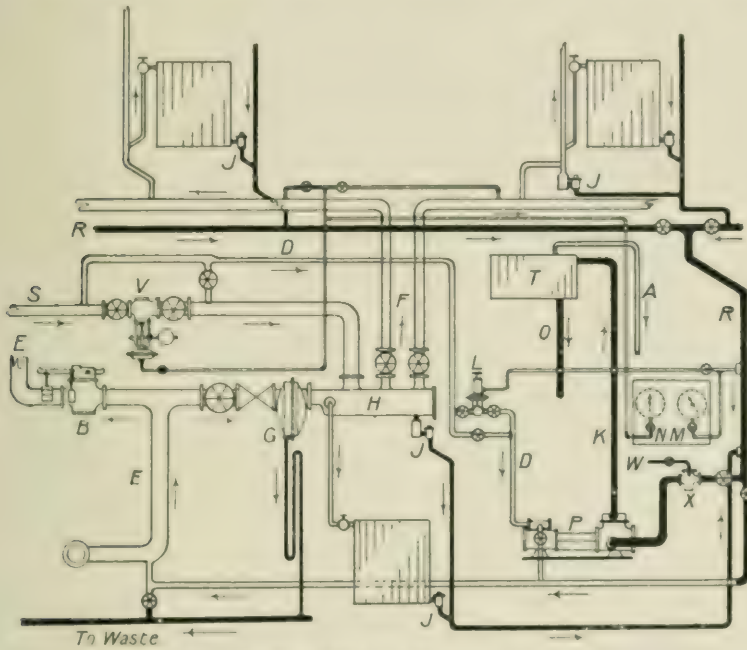


FIG. 119.—The "Webster" vacuum system.

J = radiator outlet valve.
R = condensation pipe.
D = steam supply to vacuum pump.
S = live steam supply.
V = pressure reducing valves.
F = steam pipes to heating systems.
T = air and condensation separating tank.
A = air escape.
O = pipe to feed-water heater
E = exhaust pipe.
B = back pressure valve.
G = grease separator.
H = steam header.
L = automatic pump governor.
K = outlet pipe from pump.
N.M = differential pressure gauge.
W = water supply.
X = grit box and strainer.

designed for the use of both "live" and "exhaust" steam. When "exhaust" steam is available, it is not necessary to have much load on the back pressure valve, and ¼ lb. per square inch should be ample. The exhaust pipe from the engine is

indicated by E, the steam to the heating plant being first passed through the grease separator G, from which it flows to the header H, being thence distributed to the points desired. At the ends of the steam mains and at low intermediate points, are introduced drip pockets, which are joined with the return mains through automatic valves. To separate the air from the water of condensation, the discharge from the vacuum pump is delivered to the overhead tank T, from which the condensation may gravitate to low-pressure boilers. If, on the other hand, the water of condensation requires to be returned to high-pressure boilers, it may be accomplished by either a feed pump or by a "return trap." For maintaining a given vacuum, an automatic valve L is employed, whereby the steam supply to the pump is controlled.

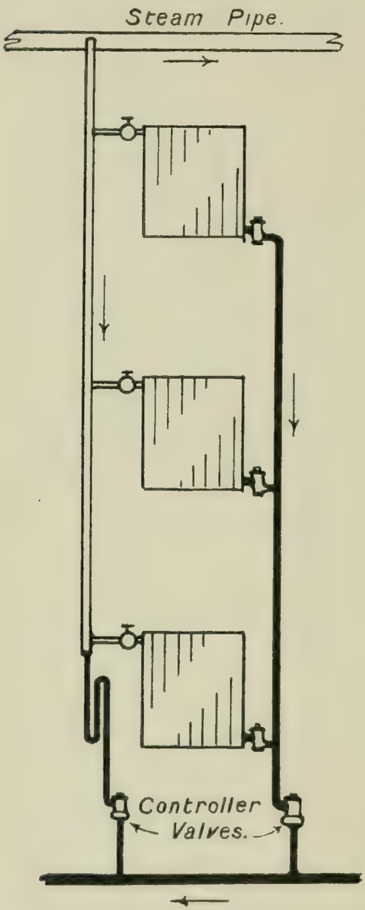


Fig. 120.—Morgan-Clark vacuum system.

"**Nunomatic**" **System.**—To a great extent, this resembles the "Webster" system, the chief difference being that, whereas in Fig. 119, both inlet and outlet regulation are used in connection with the heating surfaces, only inlet regulation is adopted in the "Nunomatic" system, the return connections being open to the return mains. Further, in order to assist the flow of condensation, the most distant return connection is joined with the main return nearest to the pump, whilst the nearest return connection is arranged to come at the

end of the return line. This method of treating the returns may in some cases use more piping than the general practice, and is indicated in Figs. 61 and 117.

"**Morgan-Clark**" **System.**—In general design, this also resembles the "Webster" system, but the outlet regulating appliances of the heating surfaces are of different construction. A controlling valve is also used in the "Morgan-Clark" system in connection with each branch return. The purpose of the latter appliance is to regulate the degree of vacuum on the returns nearest to the pump, and to maintain a higher degree of vacuum at the more distant returns. The application of this method of outlet regulation is shown in Fig. 120, a section of the appliance being illustrated later.

"**Sure Seal**" **System.**—Yet another system in which the air

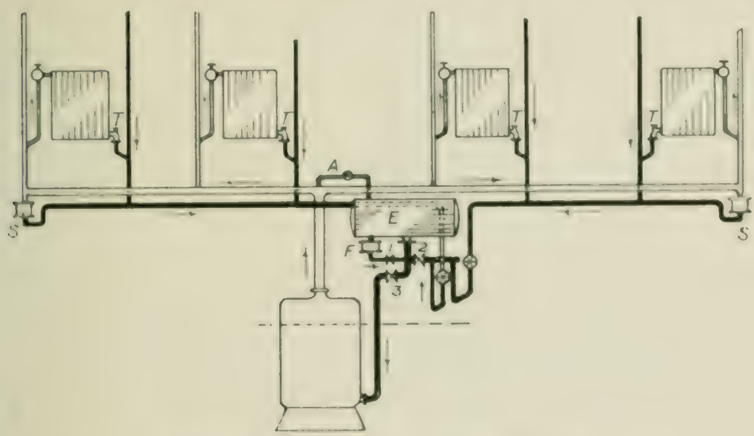


FIG. 121.—"Sure seal" vacuum system.

S = water seal.
A = steam to exhauster.
E = exhauster.
F = float chamber for operating air and steam valve.
T = sealed outlet.

and condensation flow together is given in Fig. 121. This possesses a few points of interest in that the automatic appliances consist mainly of mechanically operated valves. To the return end of each radiator a special fitting is attached, whilst for joining a "drip" with a return pipe a float-controlled valve is employed; the latter device is responsible for the enlarged

"drip" pockets at the bottom of the steam risers on the right and left of the figure. For creating the vacuum, a condenser is used, and each return main is trapped before being connected with the condenser. As in the "Dunham" system, no condensing water is used.

CHAPTER XIV

ACCESSORIES FOR VACUUM SYSTEMS

Radiator Valves.—To a great extent the form of radiator valve is dependent upon the design of the plant. For example, where the air and condensation travel together, graduated or fractional valves may be used for regulating the steam supply. On the

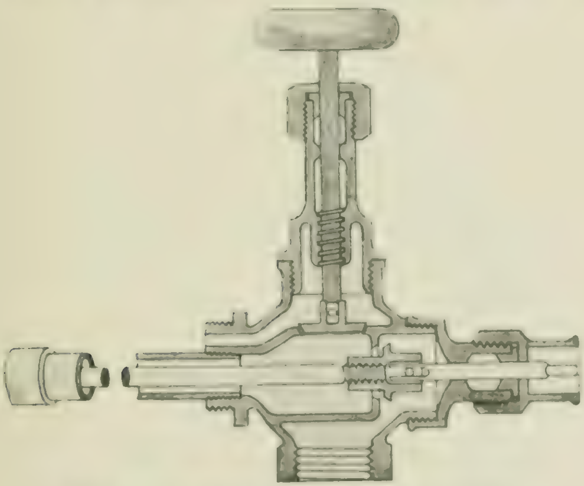


FIG. 122. Ashwell and Nesbit's radiator valve.

other hand, in installations where independent air lines are used, full-way valves are the more suitable. In either case, it is desirable to use a good type so as to avoid the infiltration of air from these points. In some systems special valves are required, one of these being indicated in Fig. 122, and this is used in the "Nunomatic" system. The appliance is attached at the bottom of a radiator, and the tube with its contained rod

projects inside. By means of the auxiliary cock on the right of Fig. 122, permanent regulation of the steam supply can be obtained to suit any particular size of radiator, whilst the upper cock provides the occupant of a room with the usual means of adjustment. The tube with its contained rod is intended to afford the necessary differential expansion for cutting off the supply of steam automatically when the radiator is fully charged.

Outlet Regulating Appliances for Radiators.—For controlling the outlets of radiators and of other heating surfaces, various appliances have been devised. These may be broadly classified as—

(1) Mechanical devices;

(2) Thermostatic or expanding valves;

(3) Fixed devices that contain no movable parts.

It is important, if the most economical results are to be obtained, that the appliances used must allow only a minimum of steam to escape.

Mechanical Devices.—Fig. 123 gives one of this form in

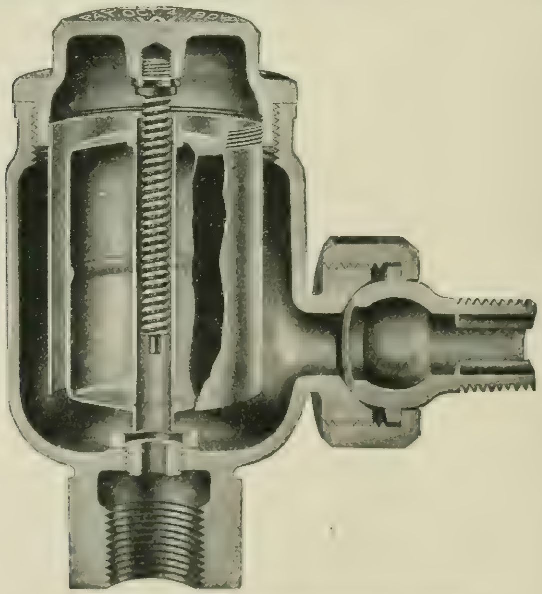


Fig. 123.—The "Webster water-seal motor."

which a float is attached to a central tube, the latter being bodily raised and opening the valve as the condensation tends to gather there. The air is drawn by the vacuum pump through the water seal and down the spiral passage of the tubular valve stem. The special feature of this device is the screwed pin that regulates the size of the air aperture.

Another mechanical type is given in Fig. 124. In this, the

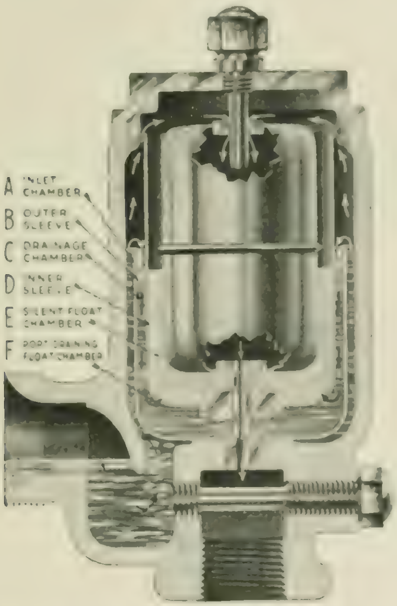


FIG. 124.—Monash raditor. By Monash-Younker Co.

air and water currents upon leaving the radiators are prevented from coming in direct contact with the float in order to prevent the chattering of the valve through this cause. The water, it will be seen, is caused to take a circuitous course to the float, whilst the air takes the passage indicated by the darts.

An entirely different form, although of the same class, is given in Fig. 125, which has a spring adjustment. In the absence of water, the valve is closed by the tension of the spring, but is opened as the condensation tends to gather and press

upon it in conjunction with the aid of the vacuum beneath. The same construction is used for the controlling valves near

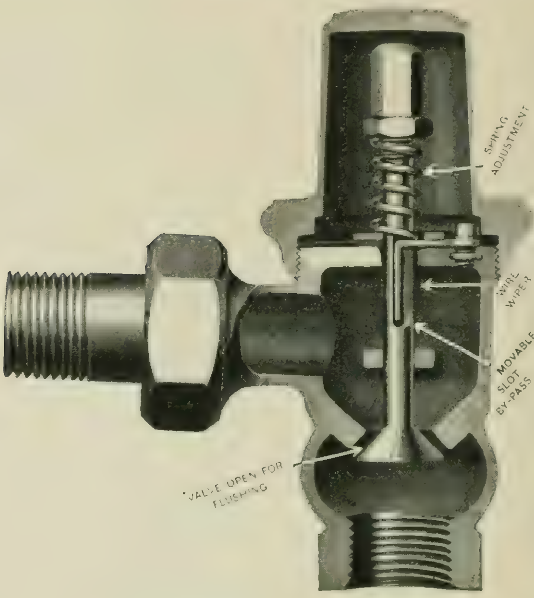


FIG. 125.—Morgan-Clark vacuum valve.

the base of the risers in Fig. 120, but these are necessarily of a larger size owing to the greater capacity required.

Thermostatic Valves.—Of these, there are many different forms, but the more sensitive ones, as a rule, make use of very volatile fluids. One form of thermostatic valve or trap is given in Fig. 126, which readily closes the outlet when in contact with steam.

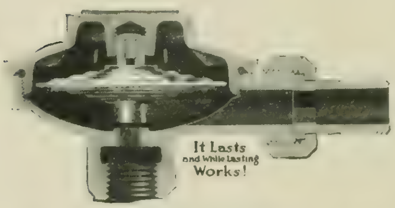


FIG. 126. The radiator trap. By the C. A. Dunham Co.

Another fitting of the same class is given on p. 124, but in this case a permanent passage for air is provided through the stem of the valve.

Fixed Devices.—A fitting of this class is used in connection with the "Sure Seal" system. By means of an inverted cone, a water seal or trap is formed at the base of the appliance, whilst the principal communication between the inlet and the outlet is through a submerged port. The idea underlying this construction is that neither the vacuum in the return, nor the usual steam pressure in the radiators will break the water seal owing to an air aperture at the upper part of the fitting which will neutralize any tendency towards differential pressure. At the same time, the water of condensation will find a ready means of escape by flowing through the submerged aperture and over the cone.

Another valve of this class largely resembles Fig. 123 with the float removed, and with the centre tube shorter and of a smaller bore. The cylinder forming the water seal is perforated above the water-level for the escape of air, and the obstruction to the flow of steam through the raised outlet nipple depends upon a globule of water being formed over it by condensation.

From the foregoing, it will be observed that there is a wide range from which to choose, but the principal factors to be considered are—

(1) Reliability of action;
(2) The percentage leakage of steam;
(3) Durability;
(4) Initial cost.

A correct knowledge on the first three points can only be gained by subjecting the different fittings to some uniform tests which should conform to actual working conditions as near as it is practicable to do so.

Exhausting Apparatus.

These take the form of pumps, condensing tanks, and injectors, each appliance having its merits and limitations. The pumps used may be either "wet" or "dry," depending upon the class of apparatus to be installed.

"Wet" Vacuum Pumps.—Where high-pressure steam is available, a directly connected steam pump is often employed, and one of this type is given in Fig. 127. The air and water

side should be provided with large valves, so as to offer as little resistance as practicable, for it is very undesirable for the

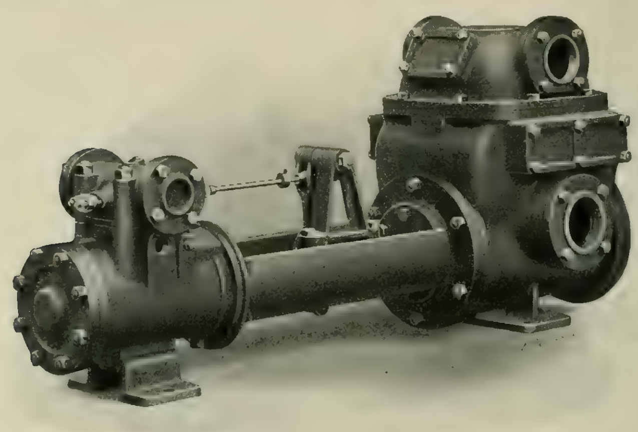


FIG. 127.—Reciprocating vacuum pump. By the Worthington Pump Co.

degree of vacuum in the pump to exceed appreciably that at the inlet. Such a condition may be responsible for the re-evaporation of the water of condensation, when the pump would "race." The valves should also be readily accessible for the purpose of overhauling and for executing repairs.

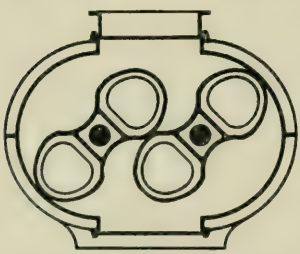


FIG. 128.—Rotary vacuum pump. By the Connersville Blower Co.

Rotary pumps are suitable where electricity is the motive power, as fairly high-speed motors may be directly joined with them. Fig. 128 gives a rotary type, the only movable parts in the casing being the two impellers shown.

Jet Water for Vacuum Pumps.—Where the piping of an installation can be favourably arranged, the use of jet water should not be necessary for maintaining the required vacuum at the pump. Where, however, a relatively high vacuum is necessary, jet water may be essential, but even then its use in many cases can be restricted by curtailing the steam supply to

the radiators. In other words, the condensation would be appreciably cooled before leaving the heating surfaces, so that its temperature at the pump would be below the boiling point corresponding to the degree of vacuum maintained there. Any good vacuum system should allow the condensation to cool in the heating surfaces by at least 8° F. below the temperature of the steam from which it is formed. The cooling of the condensation cannot be done, however, without cost, and the question may arise, when designing a plant, whether this charge shall form part of the initial outlay by increasing the area of the cooling surface and so diminishing its average temperature, or whether the operating cost should be raised to keep down the initial charge. Very often when jet water is used, much is needlessly wasted, although this may be obviated to some extent by automatically controlling the water valve.

Dry Vacuum Pumps. These are only applicable where an

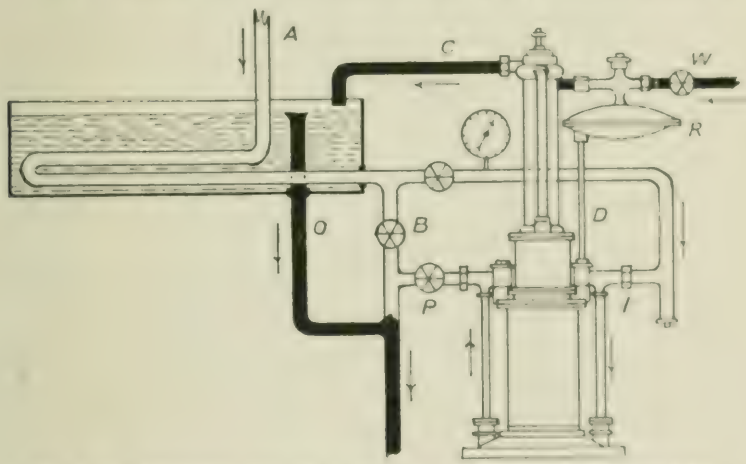


FIG. 129. Automatic hydraulic pump. By the Bishop and Babcock Co.

A = air line.
C = exhaust water from pump.
W = water supply.
R = diaphragm and valve.
B = by-pass.
D = vacuum tube.
I = inlet to pump.
P = outlet of pump.

independent air-line is used. In Fig. 129, a hydraulic pump is given, which shows the usual method of connection with the main air-line. It is a double-acting form, and is automatically

governed by a diaphragm, R, which is reacted upon by the vacuum, and which in turn operates the water-supply valve. Before joining with the pump, a length of the air-line is submerged in a shallow tank so as to condense any steam that happens to reach that point. A by-pass is also shown to enable the pump to be cut out of use when desired.

Condensing Tanks.—The chief features in connection with these exhausters are: the manner in which the steam is admitted for producing the vacuum, and the means adopted for discharging air.

Injectors.—These appliances have the advantage of occupying little space; they are also simple and comparatively cheap.

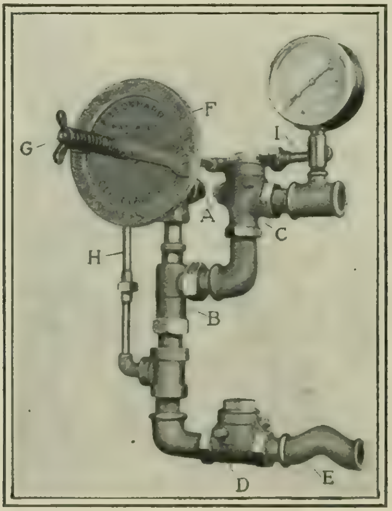


FIG. 130.—Automatic injector. By F. Leonhard, Cleveland, O.

F = diaphragm.
G = vacuum adjustment screw.
I = vacuum pipe joining with diaphragm.
A = water-supply connection.
B = ejector.
C = air pipe to ejector.
H = small water escape pipe from cylinder behind diaphragm.
D = check valve.
E = air trap.

Both steam and water injectors are in use, an automatic one of the latter class being shown in Fig. 130. At A is a cylinder containing a differential plunger which controls the supply of water to the injector, whilst a small port is formed at the

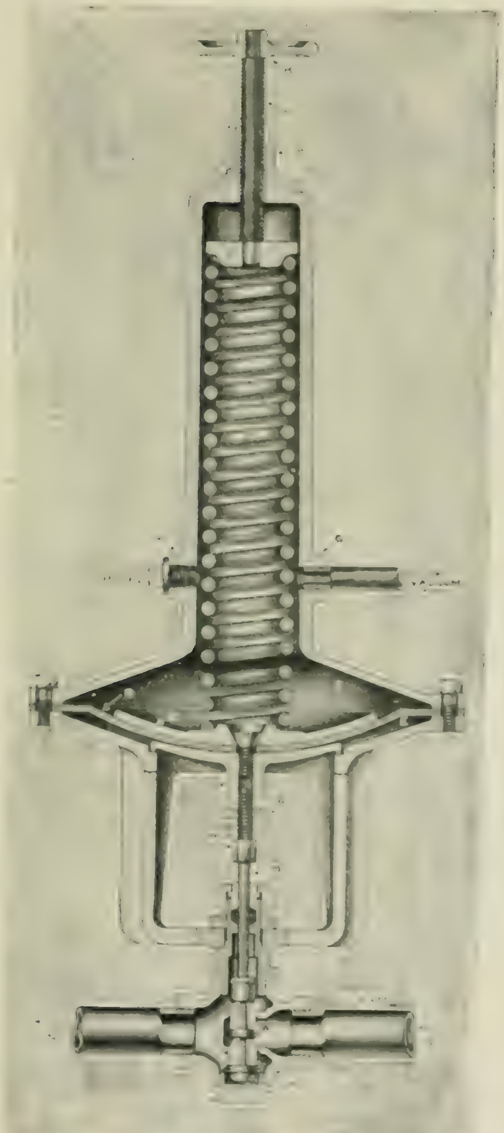


Fig. 131.—The "Webster" pump governor.

top of the cylinder which is opened and closed by the spindle G. To the same port is connected the small tube H, which drains away any water that passes by that route. At F, a vacuum chamber is formed by a metal diaphragm which communicates with the air-line through the tube I. When the degree of vacuum is below the desired amount, a spring on the diaphragm spindle keeps the small port open, when the water flows to the injector and withdraws the air from the system. If, however, the vacuum tends to rise beyond the regulated amount, the diaphragm is pressed inwards by the atmospheric pressure closing the relief port, and in turn concentrating the pressure on the differential plunger and cutting off the water supply. The degree of vacuum is adjusted by the wing nut on the spindle G.

The use of injectors is chiefly confined to systems in which independent air-lines are used, but they are not so economical in operation as pumps for large plants.

Tank exhausters have a large field of usefulness, and are more economical than either injectors or pumps. The latter, however, are more suitable for large installations, and for those in which a continuous and positive action is required.

Automatic Regulation for Vacuum Pumps.—Reference has already been made to the regulation of vacuum pumps, as well as to some of the means by which it may be effected. With either hydraulic or steam driven pumps, the degree of vacuum may be regulated to a nicety by controlling the water or steam supply. With electrically driven pumps, the speed cannot be adjusted to the same degree, for it is either necessary to stop them completely when the vacuum rises too high, or to run them at a certain speed, whilst the degree of vacuum is often adjusted by admitting the inflow of some air. Where the latter practice is adopted, it is desirable to be able to run the motor at different speeds in order to cut down the operating cost when there is only a small heating load. Fig. 131 gives a steam-pump governor which is fixed in the steam supply pipe. The valve, it will be seen, is of the double-seated form, and is closed by the diaphragm when reacted upon by the vacuum in the return pipe. By the hand-wheel at the top of the fitting, it can be set to suit any vacuum desired.

CHAPTER XV

HEATING SURFACES

THE heat emitted from surfaces for warming purposes is chiefly of two kinds, viz. radiant and convected. Both forms of heat transfer are operative in the case of direct or exposed heaters, whilst obscured or indirect heaters warm mainly by convection.

Radiant Heat.—The quantity of heat radiated from a surface depends principally upon the following:—

(*a*) Temperature of air-enveloping heater.

(*b*) Mean difference in temperature between the surrounding air and the heater surface.

(*c*) The material of which the heater is made and its surface condition.

(*d*) The manner in which the surfaces are formed and grouped.

(*e*) The position in which the surfaces are placed.

Radiant heat is transmitted or projected in straight lines from the radiant to any object or recipient at a lower temperature.

It is often assumed that the transmission factor of exposed heating surfaces is fairly uniform, but this only applies to a very small temperature range, the value increasing both with respect to radiant and convected heat, as the temperature difference between the heater and recipient increases. For example, the transmission factor for radiant heat would be approximately 30 per cent. greater for a temperature difference of 160° than for a difference of 45° F.

Effect of Heater Surface.—The material of which a heater is made affects the radiant heat value to a considerable extent, so long as its outer surface remains in a natural or untreated state. As a rule, metals are used for heating surfaces, but porcelain and concrete radiators are also made. If plain cast iron is adopted

as a standard material, and its transmission factor for radiant heat is taken as unity or 1, that for silvered-plated copper surfaces would be approximately 0·041; for plain copper, about 0·05, and for wrought iron 0·87. Now, although these values differ so much, the enamelling or painting of the different surfaces would make their radiant heat transmission factors nearly alike. This is an economical feature of which advantage is not always taken.

The results of a very interesting series of experiments upon the heat transfer from radiators when coated with different preparations, were given by Prof. John R. Allen in a paper to the National District Heating Association in 1911. In Prof. Allen's experiments, two similar radiators were used in order to obtain his results, the one being coated with the preparations under test, whilst the other was left plain. Upon painting the test radiator with two coats of copper bronze, its efficiency was observed to fall 24 per cent., and by a further 1 per cent. when two additional coatings of the same substance were applied. Two coats of terra-cotta enamel were afterwards added, when the efficiency of the radiator was raised to 3 per cent. above the plain or checking one. By the application of further coatings, the efficiency of the radiator was lowered or raised according to the preparation used. The results of these tests coincide with others independently obtained, and show that the bronze powders or preparations reduce the heat-transmitting capacity of radiators, whilst enamels, varnishes, lamp black, and oil paints with a lead or zinc base tend rather to raise it.

Effect of Grouping.—The arrangement and grouping of the heating surfaces affect the radiant heat transmitted, in that a more or less percentage is rendered ineffective through being simply re-radiated from surface to surface; in other words, by being unable to get away. The greatest percentage of radiant heat is given off by single lines of pipes when freely exposed, the value diminishing when they are grouped together, and according to their distance apart. For these reasons, single-column radiators are better than those with a greater number, but two or more columns are often necessary for concentrating the heating surface into a limited space.

Convected Heat.—This is mainly influenced by the following conditions:—

1. The difference in temperature between the heater and the air in its vicinity.
2. The height and form of the surfaces.
3. The velocity of the air passing over the surfaces.
4. The degree of contact between the heater and recipient.
5. The location of the heating surfaces.

According to the researches of Dulong, the heat transfer by convection from any given surface in still air is directly proportional to the 1·233 power of the temperature difference, multiplied by a "constant," and by a transmission value which varies with the surface under consideration. His deductions appear to be in close agreement with facts, for computed values differ very little from those obtained by experiment.

The greatest quantity of heat emitted by convection per unit area of surface occurs when the latter takes the form of small-bore horizontal pipes. If the pipes are vertically arranged, the rate of heat transfer gradually diminishes from the bottom owing to the reduced temperature difference at the higher levels. Low radiators, therefore, are more effective than high ones, other conditions being equal. Unlike radiant heat, convected heat does not appear to be influenced to any appreciable extent by the surface condition of the heater.

For the transmission values of radiant and convected heat, the reader is referred to the experiments of Péclet, results of which are given in Box's "Treatise of Heat."

Effect of Air Velocity.—With respect to the velocity of air over a heating surface, the transmission coefficient increases with increased velocity, but the rise of the value does not appear to follow any definite law. Experimental values are therefore necessary for changes of air velocity.

Resistances to the Transfer of Heat.—There are three forms of resistances which affect the passage of heat from a radiator or similar surface. The first is the inner film which opposes its entry into the material. When water is the heating medium, this resistance is negligible, and it is only of small moment when saturated steam is used. On the other hand, when steam enters a radiator with a large degree of super-heat, a more or less

percentage of its inner surface remains unwetted, and in consequence the rate of heat transfer by the metal is diminished. The second resistance is the thickness of the material of which the heater is made. This, however, in the case of metals, is of no great importance. Under general circumstances, it is the outer surface that offers the greatest resistance to the transference of heat. This has already been clearly shown, for by the mere coating of a radiator with different preparations, its capacity for transmitting heat may be affected to a very appreciable extent.

Direct Heating Surfaces.—For the effective distribution of heat in an apartment, pipes as heating surfaces are probably the best, but they have the drawbacks of inconvenience and unsightliness, especially when the heating agent is circulated at a low temperature. They are very serviceable, however, in many structures, such as works, etc., where appearance is of secondary importance.

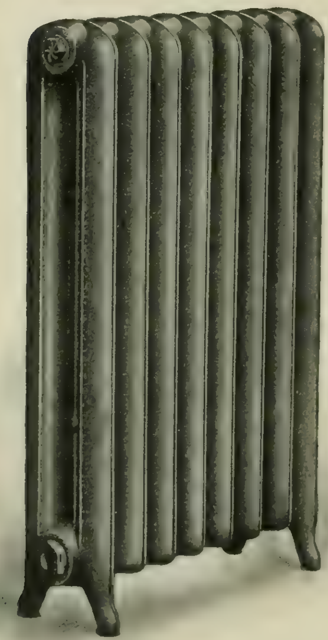


FIG. 132. Two-column radiator. By National Radiator Co., Ltd.

Radiators.—There are numerous forms and varying heights of radiators, and they may take either a plain or an ornamental style. It is well, when locating these in rooms, to keep the units as small as practicable, for much better results are obtained with the total surface well distributed. Whether steam or hot water be the heating medium, the columns of radiators should be joined with screwed nipples both at the top

and the bottom. Steam radiators sometimes have their columns joined only at the base, but the practice is a bad one owing to the difficulty there is in displacing air from them.

Fig. 133. Radiator bushing. By Lumley, Son, & Wood, Ltd.

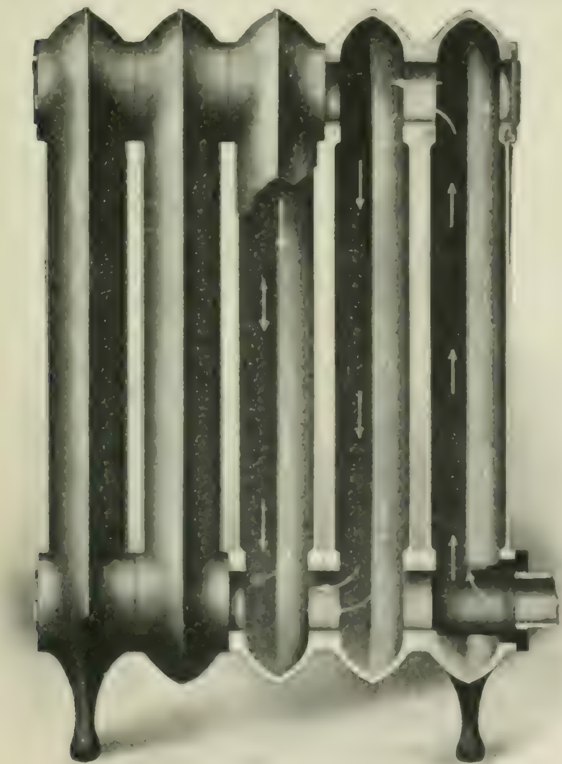


Fig. 134. Showing application of radiator bushing. By Lumley, Son, & Wood, Ltd.

From a sanitary standpoint, ornamental radiators are not desirable, as the raised parts collect dust, and they become more objectionable still when high temperatures are carried.

Fig. 132 gives a plain two-column radiator, each 38-inch

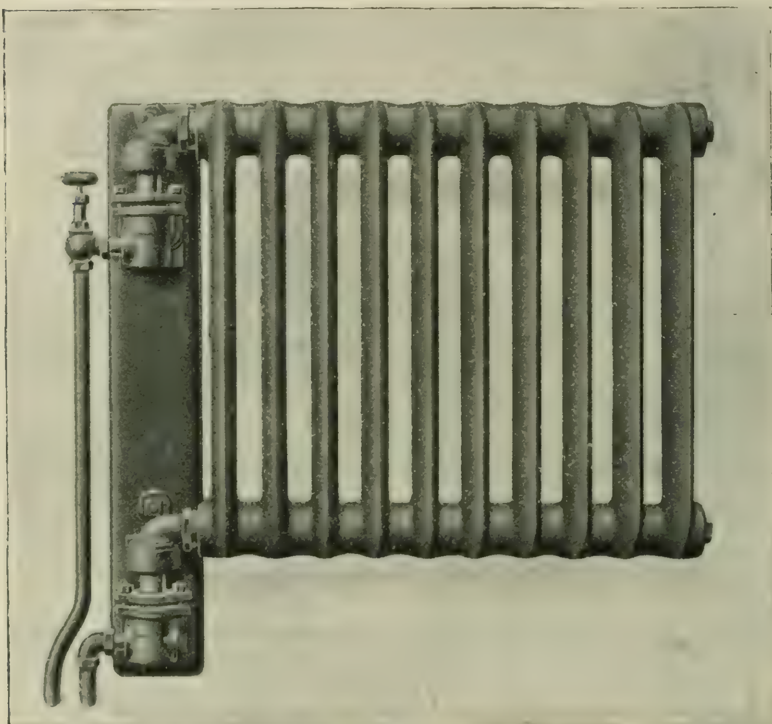


FIG. 135.—Swing radiator. By Beeston Foundry Co.

section being listed as containing 4 square feet, and 2 square feet for a height of 20 inches. It has already been shown that the lower radiators are more effective per unit area of surface so far as heat transfer is concerned, but the higher ones have the economical advantage of costing less per square foot of surface.

A convenient form of radiator "bushing" is given in Fig. 133, whilst its application is shown in Fig. 134. It will be observed, that, the end of the bushing being sealed, the first

column of the radiator acts as a continued branch supply, or as the flow column.

Fig. 135 gives a useful radiator which is provided with gland joints, thus enabling it to be swung clear of the walls for cleansing or other purposes. It is chiefly intended for hospital use, and has single columns 7 inches wide and spaced $1\frac{3}{4}$ inches apart. For a height of 36 inches, each section is listed as containing $3\frac{3}{4}$ square feet of surface, and 3 square feet for 30 inches high.

Humidifying Radiators.—When it is desired to add additional moisture to the atmosphere of a room, evaporating pans are often attached to the heating surfaces. The pans take

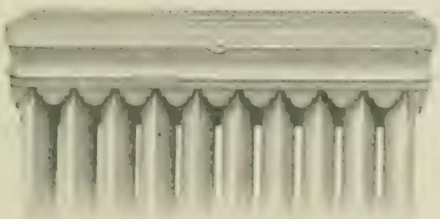


FIG. 136. Radiator with humidifying pan. By National Radiator Co.

different forms, and may be arranged either to sit on the top of the radiators, as in Fig. 136, or small independent pans may be located between the columns. In the latter case, more surface can be provided to come in contact with the radiators, and thus the rate of evaporation is increased.

Porcelain radiators are sometimes used for increasing the humidity as well as for heating, and for the former, they have met with some degree of success when steam is circulated through them. Drawbacks have arisen, however, on account of the uncertainty of the material, and the difficulty of making the connections tight, whilst drip pans are often necessary beneath them to intercept leakage and surplus moisture. In Germany, concrete radiators have been constructed with the same object in view, but as to their efficacy the writer has no first-hand knowledge at the time of writing.

Radiator Shields.—In cases where radiators are placed close to walls, the latter get soiled immediately over the heating surfaces owing to the particles of dust being projected against them. This is specially marked with highly heated surfaces, and to avoid it, radiators are sometimes provided with shields, as in Fig. 137. The shields are of varied designs, and are made either for deflecting the dust from the walls or for intercepting it.

Location of Radiators.—For a long time, it was simply

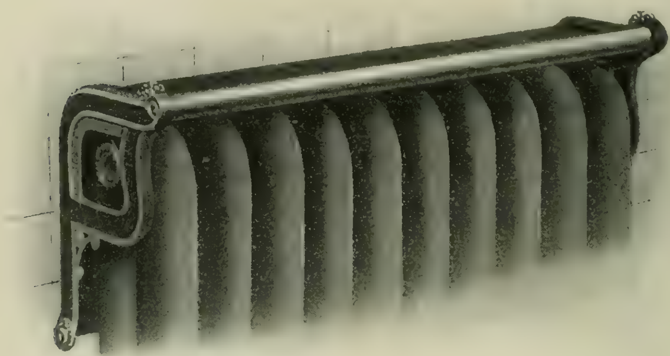


FIG. 137.—Radiator with shield. By Lumley, Son, & Wood, Ltd.

assumed that the best positions for locating radiators in rooms were those most exposed, such as beneath windows and near external walls. More recently, however, this matter has received greater attention, and from observations, it is found that a better efficiency and a more effective diffusion of the air currents can be obtained by placing them near internal walls.

When radiators are fixed beneath windows the air currents set in motion do not as a rule penetrate far into a room, but have a tendency to "short circuit," returning along the floor. This is especially marked when there is much inward leakage of cold air from the windows. On the other hand, by locating radiators against the warmer internal walls, there is less loss of heat by conduction, whilst the air freely circulates from the warmer to the cooler surfaces in accordance with natural law.

Cause of Draughtiness.—In buildings with large exposed cooling surfaces, considerable discomfort may be experienced by draughts. When a rising column of warm air is brought in contact with cold clammy surfaces, it is rapidly cooled, and then falls as a draught upon the people beneath. This is liable to occur in buildings that are irregularly heated, and where the walls do not get properly warmed. To avoid the trouble, various expedients are resorted to, such as the provision of double glazing, locating heating surfaces behind screens so as to deflect and concentrate warm air currents upon the cold surfaces. The upper portions of high walls and roofs are also separately warmed by placing heating surfaces at those points. The heating of the upper exposed surfaces is specially important in buildings where considerable moisture is present, in order to obviate trouble through condensation.

CHAPTER XVI

VENTILATING AND INDIRECT RADIATORS

Ventilating Radiators.—A common practice of warming the inflowing air to an apartment is shown in Fig. 138, where baffle plates are fitted to the radiators. The deflecting plates are secured in various ways, such as by screws, springs, and lugs, but the method adopted should be a simple one, so that

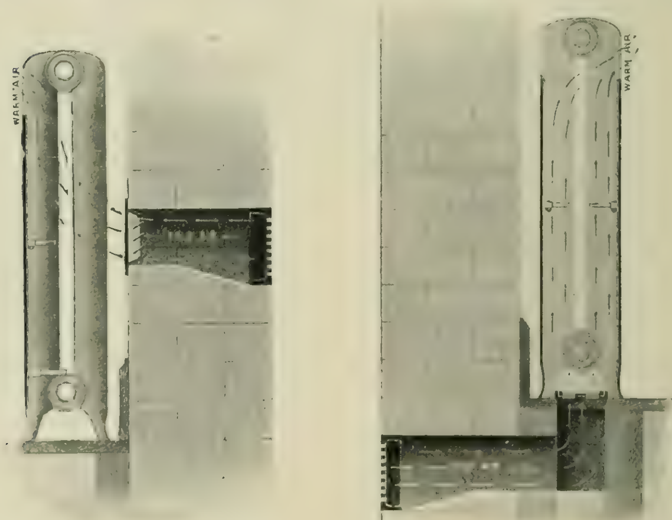


Fig. 138.—Ventilating radiator. By National Radiator Co.

the plates can be readily detached for the cleansing of the surfaces.

Fig. 139 gives a flue type of ventilating radiator, and with a suitable air supply this is fairly effective, for it is designed to

bring the air in contact with the surfaces as well as to provide a fair percentage of direct surface for the emission of radiant heat. The chief drawbacks associated with flue radiators arise through their hidden surfaces, and the difficulty of keeping them clean, although the latter defect is to a great extent avoided in the figure shown.

All ventilating radiators should be provided with some simple and effective means for regulating the air supply. In Fig. 138, the air inlet of the one is controlled at the base by a slotted slide, and that of the other with a louvred ventilator which is operated by a lever and quadrant above the radiator.

Sizes of Ducts to Ventilating Radiators.—When only short inlet ducts are used, as in Figs. 138 and 139, and the exhaust outlets extend to the top of a building, the air velocity through the inlets may be taken as 3 to 5 feet per second, according to the extracting power of the outlet shafts. On the other hand, where no regular outlet ventilation is provided, the flow of air into a building cannot be ascertained with any degree of precision.

Assuming the air velocity is arbitrarily fixed, the duct area can be obtained by the following simple rule:

$$a = \frac{\cdot 04Q}{v} \quad . \quad . \quad . \quad (21)$$

and

$$Q = 25av \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

where Q = volume of air in cubic feet per hour.

a = area of duct in square inches.

v = velocity of air in feet per second.

It is desirable, where possible, to provide at least 1800 cubic feet per occupant per hour, and to distribute so as to avoid unpleasant draughts. Generally speaking, when cold air enters an apartment, not more than three air changes per hour can be effected without draughts being felt, but if the air is tempered with ventilating radiators, double the above should be obtainable without causing discomfort. The total area of the inlet ducts may be made up of any given number of smaller channels, and these may be of equal or of different sizes.

The capacity of ventilating radiators is limited for handling air, as the following example will show.

Example 16.—What volume of air would enter a large room per hour through four 6″ × 18″ registers behind radiators,

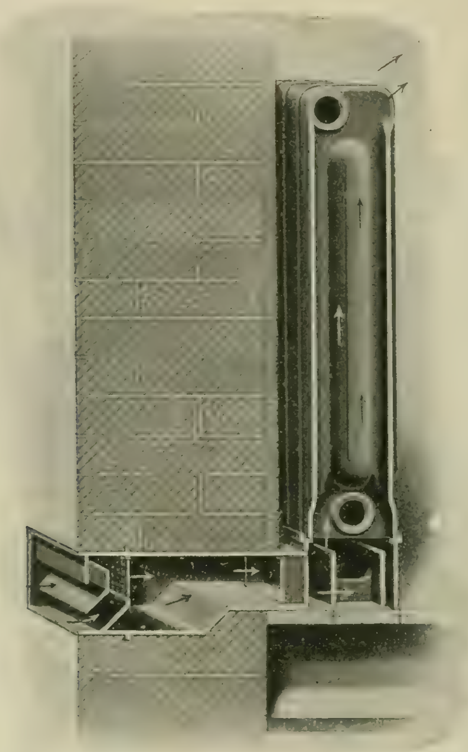


FIG. 139.—Ventilating flue radiator. By Beeston Foundry Co.

if the net air space is 65 per cent. of the grate surface and the air velocity is 4 feet per second?

For this case, the total clear air space will be

$$4 \times 6 \times 18 \times 0{\cdot}65 = 280{\cdot}8 \text{ square inches.}$$

By formula 22—

$$Q = 25av.$$

Substituting values, $Q = 25 \times 280{\cdot}8 \times 4.$

When $Q =$ **28,080 cubic feet per hour.**

Assuming this room were occupied by sixty persons, the air supply per individual per hour would be $\frac{28,080}{60} = 468$ **cubic feet**, or about one-fourth the volume desirable.

Indirect Heaters.—Wrought-iron pipes have been largely used for these, although within recent years cast-iron heaters have been adopted. When wrought-iron coils are used and steam is the heating medium, they should be formed from short rather than from long lengths of tube. Long continuous coils are never effectively drained of the condensation when the temperature difference is great, owing to a partial vacuum occurring towards the outlet end of the tube; neither is the heating surface as effective as it should be.

As far as it is practicable, the ducts supplying fresh air to indirect heaters should be formed in interior walls, and should be readily accessible. Fig. 140 gives one type of indirect

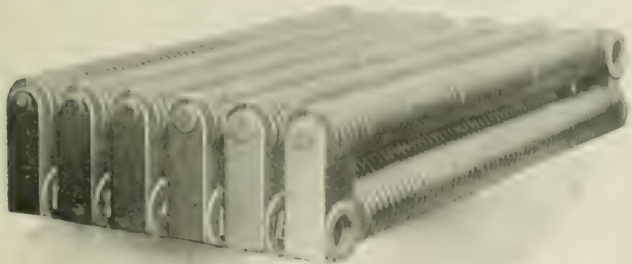


FIG. 140.—Indirect heater. By the National Radiator Co., Ltd.

heater, and Fig. 141 shows how it is sometimes arranged for warming the cold inflowing air. If one section is inadequate to raise the air to the temperature desired, another may be added to increase the depth of heater.

When air inlets are formed in buildings during their erection, and where two or more branch ducts can be supplied from one common source, the method indicated in Fig. 142 may be adopted. In this case, the lower portion of the duct is assumed to terminate in a passage or basement which can be used as a fresh-air chamber. To the latter, the external air may be

admitted through windows, louvres, or other openings, whilst screens may be arranged to arrest the larger particles of matter in suspension.

Fig. 143 gives another indirect heater, and although this was originally designed for use with fans, it is also suitable for natural draught. It is of good design, and the area of the

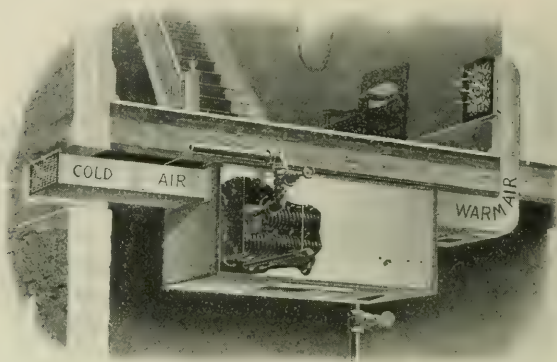


FIG. 141.

heating surface is liberal when compared with the free air space between the sections.

Another method of indirect heating is that where the coils or radiators are placed against external walls and behind wainscoting or other construction, and where a short air duct communicates directly with the external air. The latter practice, as generally carried out, is not a good one, for neither is the air well distributed throughout an apartment, nor are the heaters very effective.

With the heaters shown in Figs. 142 and 143 it is necessary to regulate the air velocity through them, as this will change with the varying atmospheric conditions. For this purpose, a simple hand-operated slide may be used when automatic appliances are not desired, whilst the ducts above the heaters may be rendered accessible by doors in the sheet-iron casings.

The most agreeable conditions in indirect heating are obtained when the temperature of the air entering a room is

kept low, but the latter factor is often controlled by the economical aspect of the problem, and by the severity of the climate where an installation is required. In Great Britain, for example, the entering air need not often be heated to over

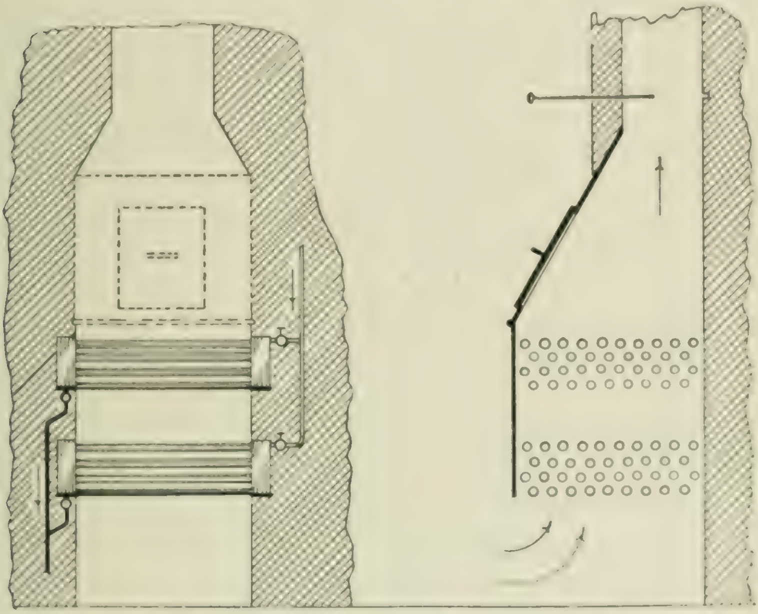


FIG. 142.—Indirect heater. Hand-controlled arrangement, where air can be by-passed through opening in sheet-iron casing when desired.

80° F., whilst in American practice it is often raised over 110° F.

Volume of Air for Gravity Indirect Heating.—The first procedure is to ascertain the total heat losses from a room as outlined in the following chapter, and then to estimate the volume of air that will yield an equivalent quantity of heat by falling through a certain temperature range. For the latter purpose formula 23 may be used—

$$Q = \frac{U}{ws(T_a - T)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

from which

$$T_a = \frac{U}{wsQ} + T \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

The total heat to be added to the cold entering air is obtained by the formula—

$$U_a = Qws(T_a - t) \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

where Q = volume of air in cubic feet per hour to warm an apartment or building.

U = total heat lost per hour by apartment.

U_a = total heat to be added to the external air.

w = density of air per cubic foot.

s = specific heat of air.

T_a = temperature of heated air entering apartment.

T = temperature to be maintained in apartment.

t = temperature of external air.

In indirect heating, there are three principal temperatures

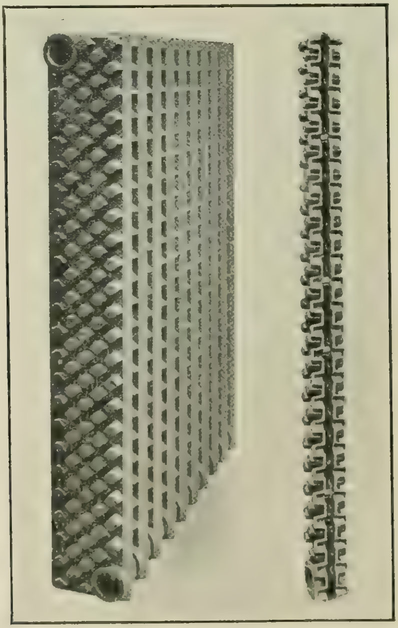


FIG. 143.—Cast-iron "Vento" heater. By National Radiator Co., Ltd.

with which one is concerned, viz. that of the heated air, the average of the apartment warmed, and that of the external air. As it is desirable to adopt a standard temperature to fix the

weight of air and to simplify the formulæ, 70° F. are chosen for this purpose. The same value is largely used in American practice, and for the sake of uniformity is adopted here. The volume of air, therefore, in passing through a duct is assumed as measured at 70° F., and if the corrected volume for any other temperature is desired, this can be readily obtained.

Assuming now that in formulæ 23 to 25 w is taken as 0·075 lb. per cubic foot, and s as 0·238, the following simple rules are derived.

Formulæ where air is measured at 70° F.—

$$Q = \frac{56U}{T_a - T} \quad . \quad . \quad . \quad . \quad . \quad (26)$$

$$T_a = \frac{56U}{Q} + T \quad . \quad . \quad . \quad . \quad . \quad (27)$$

$$U_a = \frac{Q(T_a - t)}{56} \quad . \quad . \quad . \quad . \quad . \quad (28)$$

The symbols have the same meaning as before. Suppose now that the volume of air at any other temperature is required when formulæ 26 to 28 are used, the corrected volume is obtained where change of barometric pressure is ignored by the rule—

$$Q_v = \frac{Q(460 + t_v)}{530} \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

where Q_v = volume in cubic feet at temperature desired.
Q = volume in cubic feet at 70° F.
t_v = temperature corresponding to Q_v.

The following will help to make the matter clear.

Example 17.—If 18,000 cubic feet of air enter a room per hour, and the total heat lost by the cooling agencies is 16,000 B.Th.U. per hour, determine—

(*a*) The required temperature of the entering air to give an average internal temperature of 70° F.

(*b*) The heat absorbed by the entering air when the external air is zero.

(*c*) The volume of cold air flowing to the heater, and of that leaving the registers of the apartment.

The first part (*a*) is obtained by formula 27—

where $$T_a = \frac{56U}{Q} + T.$$

Substituting values—

$$T_a = \left(\frac{56 \times 16{,}000}{18{,}000}\right) + 70;$$

when $$T_a = \mathbf{119\tfrac{7}{9}} \text{, or say } \mathbf{120° F}.$$

The second portion (*b*) can be found by formula 28—

where $$U_a = \frac{Q(T_a - t)}{56}.$$

Substituting values—

$$U_a = \frac{18{,}000 \times (120 - 0)}{56};$$

when $$U_a = \text{say } \mathbf{38{,}560\ B.Th.U.}$$

The part (*c*) may be solved by formula 29—

where $$Q_v = \frac{Q(460 + t_v)}{530}.$$

Substituting values for air at zero—

$$Q_v = \frac{18{,}000 \times (460 + 0)}{530};$$

when Q = say **15,620 cubic feet** flowing to heater.

Substituting temperature value at register—

$$Q_v = \frac{18{,}000 \times (460 + 120)}{530};$$

when Q_v = say **19,700 cubic feet at register.**

Example 18.—The heat lost by the cooling agencies of a room is 12,000 B.Th.U. per hour, when the external air is at 30° F., and when the average inside air is at 60° F. Taking the temperature of air at the registers as 80° F., determine—

(1) Volume of air required to maintain temperature of apartment.

(2) Heat absorbed by entering air.

The first part may be solved by formula 26—

where $$Q = \frac{56U}{T_a - T}$$

Substituting values—

$$Q = \frac{56 \times 12{,}000}{80 - 60};$$

when $Q =$ **33,600 cubic feet** measured at 70° F.

The second part is solved by formula 28—

where $$U_a = \frac{Q(T_a - t)}{56}.$$

Substituting values—

$$U_a = \frac{33{,}600 \times (80 - 30)}{56};$$

when $U_a =$ **30,000 B.Th.U.** per hour.

CHAPTER XVII

HEAT LOSSES FROM BUILDINGS

Heat is lost by conduction through windows, walls, ceilings, floors, etc., by air-outlet channels, by crevices due to poor construction, and by the diffusion of air through the materials of construction.

The volume of air that will pass through external walls depends upon their degree of porosity, and in some cases may equal twice the capacity of an apartment per hour. As a rule, however, it will be less than this, and will probably not exceed one air change per hour. Much depends upon the construction, and upon how the inner surfaces of the walls are treated. For example, oil-painted and varnished walls will lose less heat than similar ones left plain, and the mere re-papering of a wall may affect its rate of heat transmission.

Heat Transmission Coefficients.—For convenience, the heat transferred by walls and other cooling agencies is often expressed in coefficients, the unit of area, temperature, and time, being a square foot, one degree Fahrenheit, and one hour respectively. Thus a wall coefficient of 0·24 means that one square foot of its surface will transmit 0·24 B.Th.U. per hour per degree difference of temperature between the internal and external ones. The flow of heat through surfaces is influenced in different ways, such as by their nature, thickness, and degree of exposure. For example, a single exposed wall in a room will transmit more heat per unit area than where there are two or more similar surfaces. In other words, the single wall has a higher transmission coefficient. This is due to the larger percentage of radiant heat it can receive from the warmer interior walls.

The following tables give coefficients for different wall and other surfaces :—

TABLE VI.

Coefficients for Brick Walls.

Heat transmitted in British thermal units per square foot of surface per hour per degree difference of temperature. Authority, Prof. Rietschel. Values translated by Prof. Kinealy.

Thickness of wall Inches.	Outside wall; one side plastered	With additional stone face			Hollow walls with 2½ ins. air space, and plastered one side
		4 ins. thick	8 ins. thick	12 ins. thick	
4	0·49	—	—	—	—
8	0·36	0·31	0·29	0·26	0·25
12	0·28	0·25	0·23	0·21	0·21
17	0·24	0·22	0·20	0·19	0·19
21	0·21	0·19	0·18	0·17	0·16
25	0·18	0·17	0·16	0·15	0·14
29	0·16	0·15	0·14	0·13	0·13
33	0·14	—	—	—	0·12
37	0·12	—	—	—	—
1	2	3	4	5	6

TABLE VII.

Heat Transmission Coefficients of Different Kinds of Walls.

Heat lost in British thermal units per square foot of surface per hour per degree difference of temperature. Inside of walls plastered.

Thickness of wall. Inches.	Solid granite.	Hollow granite.	Concrete.	Sandstone.	Limestone.
	Authorities.				
	Prof. Homer.			Prof. Rietschel.	
4	—	—	0·6	—	—
6	—	—	0·5	—	—
8	—	—	0·42	—	—
10	—	—	0·37	—	—
12	0·4	0·31	0·34	0·45	0·49
16	—	—	0·30	0·39	0·43
18	0·34	0·27	—	0·36	—
20	—	—	—	0·35	0·38
21	0·31	0·25	—	—	—
24	0·29	0·24	—	0·31	0·35
27	0·28	0·23	—	—	—
28	—	—	—	0·28	0·31
30	0·26	0·22	—	0·27	—
32	—	—	—	0·26	0·28
33	0·25	0·21	—	—	—
36	0·23	0·20	—	0·24	0·26
40	0·22	0·19	—	0·22	0·24
44	—	—	—	0·21	0·23
48	—	—	—	0·19	0·21
1	2	3	4	5	6

TABLE VIII.

HEAT TRANSMISSION COEFFICIENTS OF GLASS AND OF OTHER SURFACES.

Heat lost in British thermal units per square foot of surface per hour per degree difference of temperature.

Sheet glass in single windows	1·0 to 1·03
Sheet glass in double windows	0·6
Double sheet glass	0·9
Plate glass	0·75
Single skylight	1·09
Double skylight	0·5
External doors	1·0
Interior doors	0·44
Single flooring and plaster ceiling	0·08
Double flooring and plaster ceiling	0·07
Single wood flooring	0·11
Double wood flooring	0·09
Lath and plaster ceiling	0·62
Stud partition lath and plastered one side	0·60
Stud partition lath and plastered both sides	0·35
Outside walls of frame buildings lath and plastered inside and boarded outside. Boards overlapping and ½ inch thick	0·56
Do. do. paper lined	0·35
Do. do. 1 inch thick	0·40
Do. do. 1½ inch thick	0·32
Do. do. 2 inches thick	0·27
Do. do. 3 inches thick	0·20
Slated roof with lathed rafters	0·8
Slated roof with boarded rafters one side	0·35
Galvanized iron roofing	1·5 to 2

The values given in Tables VI. to VIII. are chiefly conduction losses, and when the construction permits of a more or less considerable air leakage, some allowance should be made for this. In Tables VI. and VII., a difference in the coefficients for solid and hollow walls is shown, but in order for the lower values of hollow construction to hold good, it is necessary for the cavities to form nearly dead spaces for air. If on the other hand the air space is freely ventilated, instead of diminishing the heat transfer, it will tend rather to increase it, as the external air is brought much nearer to the inner surface of wall.

Heat absorbed by Air.—It has been already shown that when the air entering an apartment is dependent upon natural

laws, the volume delivered will vary from time to time. For a given case, the variability of the wind alone may affect the air supply by upwards of 200 per cent. In very high buildings, considerable infiltration of air will often occur at the lower floors through the aspirating effect of staircase wells and elevator shafts, and these factors should be taken into account. Much discretion is required in even approximating the probable supply of air, but a common allowance where no special inlets are provided for ordinary rooms is one air change per hour. On the other hand, if exhaust flues or chimneys are in use, not less than three air changes should be allowed.

The heat absorbed by air may be calculated by formula 25, and by formula 28 when the volume is measured 70° F.

Heat lost by Walls, Glass, and other Cooling Surfaces, may be expressed by the following:—

$$U_c = (A_1k_1 + A_2k_2 + A_3k_3 +)(T - t) \quad . \quad (30)$$

where U_c = heat lost in British thermal units by cooling surfaces.

$A_1A_2A_3$ = area in square feet of the surfaces under consideration.

$k_1k_2k_3$ = heat transmission coefficients agreeing with the surfaces $A_1A_2A_3$.

T = average temperature of air in contact with surfaces.

t = temperature of external air.

Formula 30 will give approximately correct results for glass and other thin materials and for walls where the temperature conditions remain unaltered for a prolonged time. External temperatures, however, vary very rapidly, and as thick walls have a considerable thermal capacity, the rate of heat transfer through these does not vary in the same way as that of thinner surfaces. There is great difficulty in ascertaining the exact rate of heat transmission through a wall at any particular time, and the thicker it is the more uncertain are the results.

Effect of Wind on Walls.—It is found that the heat transfer through walls is appreciably influenced by the velocity of the wind, but its immediate effect may not be so pronounced upon the quantity of heat required for a certain room as its indirect influence. For example, the first effect of an increased

wind velocity on thick homogeneous walls appears to be the abstraction of heat from the walls themselves, whilst afterwards the flow of heat into the walls is increased by virtue of the diminished temperature of the mass. The latter effect may even occur for a time when the wind velocity has fallen, and when the external temperature is somewhat raised.

This view of the problem may account for the maximum loss of heat from buildings not always coinciding with the precise period of the lowest external temperatures.

The height of rooms also affects the rate of heat transmission, owing to the mean temperature being raised. There is, however, some difficulty in formulating a rule that will give the average temperature for all heights and conditions of rooms. Prof. Rietschel gives the formula—

$$T_1 = \{0{\cdot}015T(H - 10)\} + T \quad . \quad . \quad . \quad (31)$$

where T_1 = average air temperature.
T = air temperature beneath breathing line.
H = height of room in feet.

It will be observed that this formula has been based for a standard height of 10 feet, but according to its author it is only applicable where T_1 does not exceed $1{\cdot}15T$.

Total Heat Losses from Rooms.—When estimating the whole of the heat lost from an apartment, it is usual to take into account the principal cooling agencies and to allow a certain percentage for the less important surfaces, also for the aspect and degree of exposure. The different heat losses may be independently estimated and tabulated, or they may be expressed by a general formula—

$$U = f(0{\cdot}016Q + Gk_1 + Wk_2 + Ak_3)(T - t) \quad . \quad (32)$$

where U = total heat lost in British thermal units per hour.
f = variable factor for height of rooms, aspect, minor losses, and heating periods.
Q = volume of air entering room per hour.
G = area of exposed glass in square feet.
W = area of exposed wall in square feet.
T = internal temperature at, say, 4 feet from floor.
t = external temperature.

A = area of any other important cooling surface.

$k_1 k_2 k_3$ = heat transmission coefficients agreeing with G, W, and A.

TABLE IX.

APPROXIMATE VALUE OF f.

Height of rooms in feet.	Temperatures	
	External 30°. Internal 60°.	External 0°. Internal 70°.
12 and under	1·05-1·4	1·05-1·4
13 to 16	1·15-1·5	1·1 -1·45
17 to 20	1·2 -1·55	1·15-1·5
1	2	3

Where provision is made for natural or gravity ventilation, the volume of air entering an apartment may be roughly estimated by the following rule:—

$$Q = 80a\sqrt{\frac{H(T - t)}{460 + t}} \quad . \quad . \quad . \quad . \quad (33)$$

where Q = volume of air in cubic feet per hour.
a = sectional area of duct in inches.
H = height of duct in feet.
T = average temperature of air in duct.
t = temperature of external air.

TABLE X.

VOLUME OF AIR IN FEET DELIVERED PER SQUARE INCH OF DUCT AREA PER HOUR.

Calculated by Formula 33.

Temp. of external air, Fahr.	0	20	40	30	40	50
Average room temp., Fahr.	70	70	70	60	60	60
Height of duct in feet.	Volume of air in cub. feet, per square inch of duct area per hour					
5	69	58	44	44	36	25
10	98	82	62	63	51	35
20	140	115	87	88	72	50
30	171	141	107	108	88	61
40	197	163	124	125	100	71
50	220	182	138	140	113	79
60	242	200	152	153	124	87
70	261	216	164	165	134	94
80	279	231	175	177	143	100
1	2	3	4	5	6	7

Example 19.—Assume the heat lost by the following is required. Room 14 feet high, southern exposure, external temperature 30° F., and internal 60° F. There is a ventilating stack 40 feet high having a cross-sectional area of 110 square inches with corresponding air-inlet duct. Take the temperature difference for estimating capacity of outlet shaft as 20° F. External wall area 210 square feet, and 70 square feet of sheet glass, their heat transmission coefficients being taken as 0·21 and 1·0 respectively.

By formula 32—

$$U = f(0{\cdot}016Q + Gk_1 + Wk_2)(T - t)$$

For a temperature difference of 20° F. when the internal temperature is 60°, the discharging capacity of the outlet duct will, according to column 6 of Table X., be 100 cubic feet per hour per square inch of section. Therefore for an area of 110 square inches the volume of air discharged will be $110 \times 100 = 11{,}000$ cubic feet. The value of f is obtained from Table IX., and for this case may be assumed as 1·15.

Substituting values—

$$U = 1{\cdot}15 \times \{(0{\cdot}016 \times 11{,}000) + (70 \times 1) + (210 \times 0{\cdot}21)\} \times (60 - 30)$$

$$U = 1{\cdot}15 \times (176 + 70 + 44) \times 30;$$

when $U =$ **say 10,000 B.Th.U.**

Formula 32 may be simplified by embodying values which represent certain conditions. Thus, if a 12-feet room that is regularly heated be considered, the wall coefficient being 0·28, glass coefficient 1·03, northern aspect, and the value of f from Table IX. be taken as 1·2 and 1·25 respectively, the following are obtained:—

For zero outside and 70° inside

$$U = 1{\cdot}34Q + 86G + 23W \quad . \quad . \quad . \quad (34)$$

For 30° outside and 60° F. inside

$$U = 0{\cdot}6Q + 39G + 10W \quad . \quad . \quad . \quad (35)$$

CHAPTER XVIII

QUANTITY OF HEAT EMITTED BY RADIATORS, PIPES, AND INDIRECT HEATERS.

Heat emitted by Direct Surfaces.—The quantity of heat transmitted by these surfaces is given in Tables XI. and XII. These values are somewhat smaller than those often given, but the writer believes they represent those usually obtained in practice. As shown in a previous chapter, many factors influence the flow of heat through surfaces, so that the values given can be only approximately correct. In much of the experimental work in connection with the transfer of heat from radiators, these surfaces have been too favourably located, and so results have been given which are not obtainable in general work.

TABLE XI.

Heat emitted in British Thermal Units per Hour per Foot Run of Wrought-Iron Pipe.

For the emission from cast-iron pipes, assume their diameters half an inch larger than the corresponding bore of wrought-iron pipes.

Bore of pipe in inches.	Mean temperature difference between heating medium and that of apartment warmed. Deg. Fahr.												
	60	70	80	90	100	110	120	130	140	150	160	170	180
	B.Th.U. emitted per foot run of pipe per hour.												
½	29	35	42	50	56	64	71	77	85	93	100	108	118
¾	34	42	50	57	65	72	80	88	97	106	115	126	136
1	40	49	58	66	75	84	93	104	114	124	134	144	156
1¼	48	56	66	77	88	100	110	122	132	146	158	174	190
1½	52	64	75	86	98	112	124	136	150	164	179	192	210
2	64	76	88	102	116	131	146	160	178	195	210	228	245
2½	74	88	104	120	136	154	172	188	205	225	247	268	290
3	87	103	122	140	160	179	200	219	240	262	287	311	340
3½	97	117	137	156	179	200	221	245	270	295	321	350	380
4	108	130	151	172	198	221	246	274	300	328	355	389	420
4½	118	142	165	190	216	242	270	298	327	358	390	425	460
5	129	157	181	210	239	269	299	330	361	395	432	470	505
6	150	180	212	245	280	315	352	390	430	470	510	560	605
1	2	3	4	5	6	7	8	9	10	11	12	13	14

TABLE XII.

HEAT EMITTED IN BRITISH THERMAL UNITS PER HOUR PER SQUARE FOOT OF RADIATOR SURFACE WHEN PLACED AGAINST EXTERNAL WALLS.

Mean difference of temp. of heating medium and that of apartment. Deg. F.	Single-column radiator.	Two-column radiator.	Three-column radiator.	Four-column radiator.
	Height of radiators, 36 inches.			
	B.Th.U. emitted per square foot surface.			
40	54	49	45	39
45	62	56	50	45
50	68	63	57	52
55	76	68	63	57
60	85	77	72	63
65	94	85	77	69
70	103	93	85	76
75	110	100	92	83
80	120	109	100	90
85	130	118	108	95
90	140	126	115	103
95	148	135	123	110
100	158	143	130	117
105	168	152	140	125
110	175	160	148	132
115	186	170	155	140
120	200	180	165	150
125	210	190	175	156
130	220	202	183	165
135	230	210	192	173
140	242	221	202	182
145	252	230	210	190
150	263	242	219	196
160	285	262	238	215
170	310	283	257	230
180	332	305	275	247
1	2	3	4	5

Heat emitted from Ventilating Radiators.—This varies with the different forms of radiators, the velocity of the air over the surfaces, and the degree of contact. These factors differ widely in the various types, so that it is not possible to give exact values. For similar types of radiators, it is often assumed, however, that their transmission values exceed those in Table XII. by 15 to 30 per cent., according to the form the air inlet takes.

Heat emitted by Gravity Indirect Heaters.—For determining

the rise of the air temperature after flowing through Indirect Heaters and the average heat emitted per square foot of surface, the following eight charts have been prepared. Those from 1 to 4 are for heaters formed of 1-inch piping, where the free air space is approximately 53 per cent. of the total face area. Each section consists of four tubes deep which are spaced with $2\frac{3}{4}$-inch centres. Charts 5 to 8 are for "Vento" Heaters with standard spacing, the free area being 44 per cent. of the total face area.

CHART 1.

Temperature to which air is warmed by indirect heaters of 1-inch diameter piping when the sections are four pipes deep. Also heat emitted per square foot of pipe surface per hour. Steam pressure, 5 lb. per square in. (equivalent temperature, 227° Fahr.). Temperature of entering air, 30° Fahr.

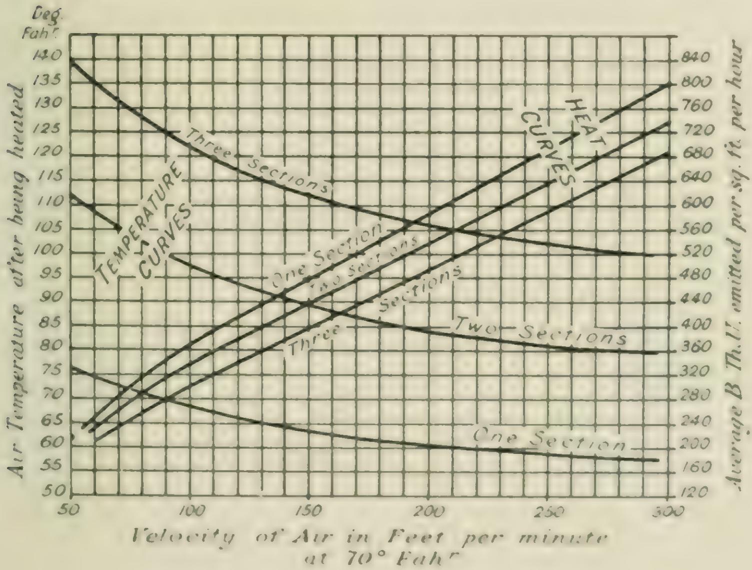


CHART 2.

Temperature to which air is warmed by indirect heaters of 1-inch diameter piping, when the sections are four pipes deep. Also heat emitted per square foot of pipe surface per hour. Average water temperature, 160° Fahr. Temperature of entering air, 30° Fahr.

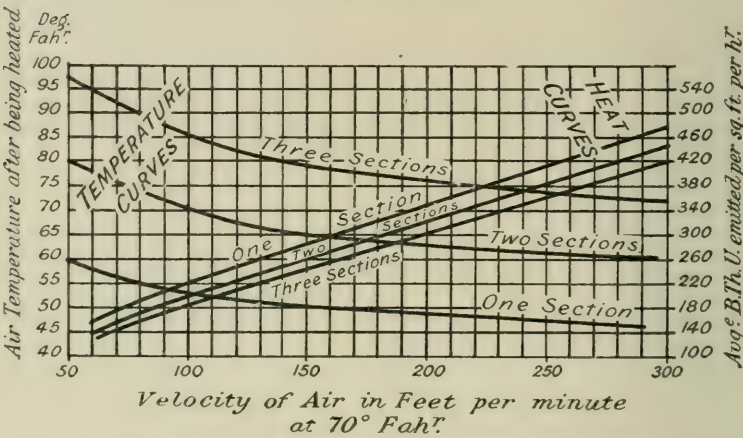


CHART 3.

Temperature to which air is warmed by indirect heaters of 1-inch diameter piping, when the sections are four pipes deep. Also heat emitted per square foot of pipe surface per hour. Steam pressure 5 lb. per square inch (equivalent temperature, 227° Fahr.). Temperature of entering air, 0° Fahr.

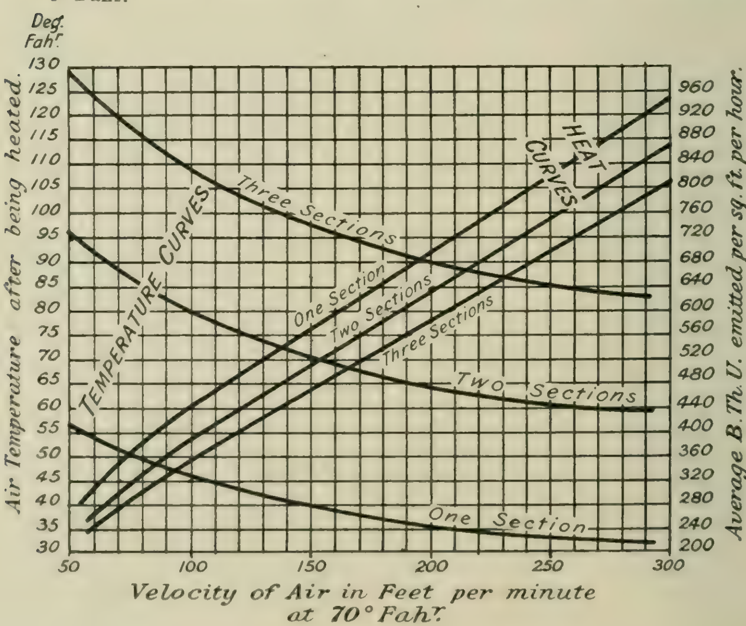


CHART 4.

Temperature to which air is warmed by indirect heaters of 1 inch diameter piping when the sections are four pipes deep. Also heat emitted per square foot of surface per hour. Average water temperature, 180° Fahr. Temperature of entering air, 0° Fahr.

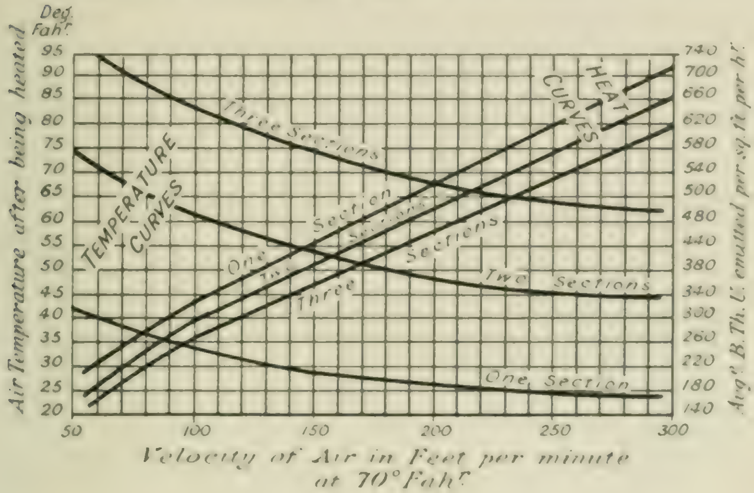


CHART 5.

Temperature to which air is warmed by "regular sections" of "Vento heaters." Also heat emitted per square foot of surface per hour. Steam pressure, 5 lb. per square inch (equivalent temperature, 227° Fahr.). Temperature of entering air, 30° Fahr. Values are taken from "Engineering Data on Vento Heaters," by the American Radiator Co.

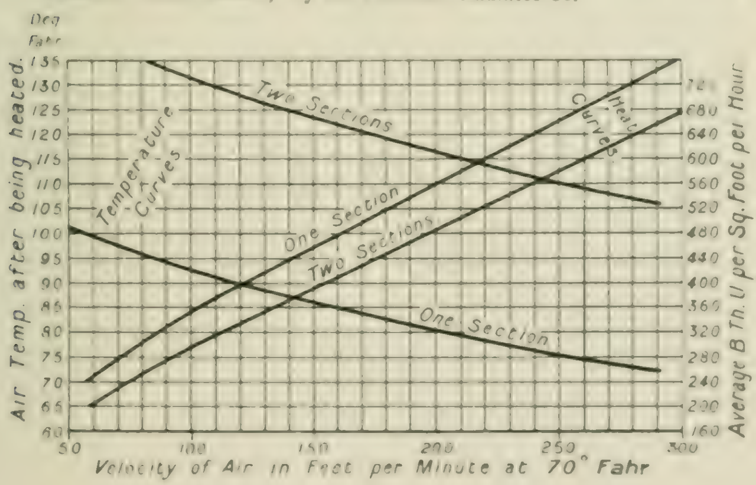


CHART 6.

Temperature to which air is warmed by "Regular Vento Heaters." Also heat emitted per square foot of surface per hour. Average water temperature, 180° Fahr. Temperature of entering air, 30° Fahr. Values taken from "Engineering Data on Vento Heaters." By the American Radiator Co.

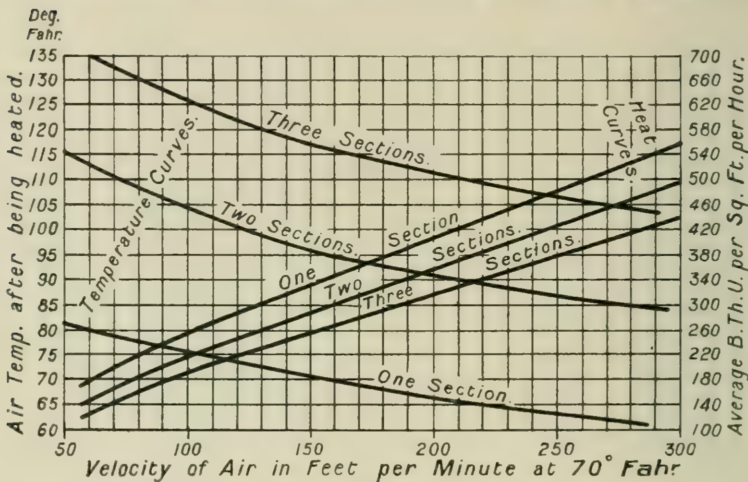


CHART 7.

Temperature to which air is warmed by "Regular Vento Heaters." Also heat emitted per square foot of surface per hour. Steam pressure, 5 lbs. per square inch (equivalent temperature, 227° Fahr.). Temperature of entering air, 0° Fahr. Values taken from "Engineering Data on Vento Heaters."

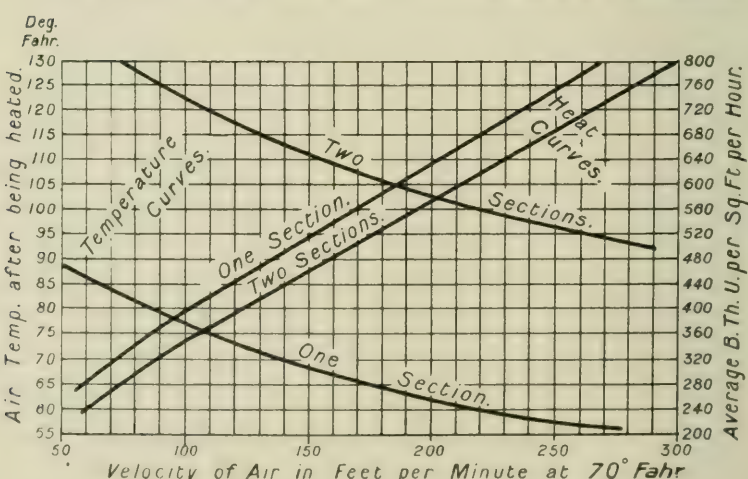
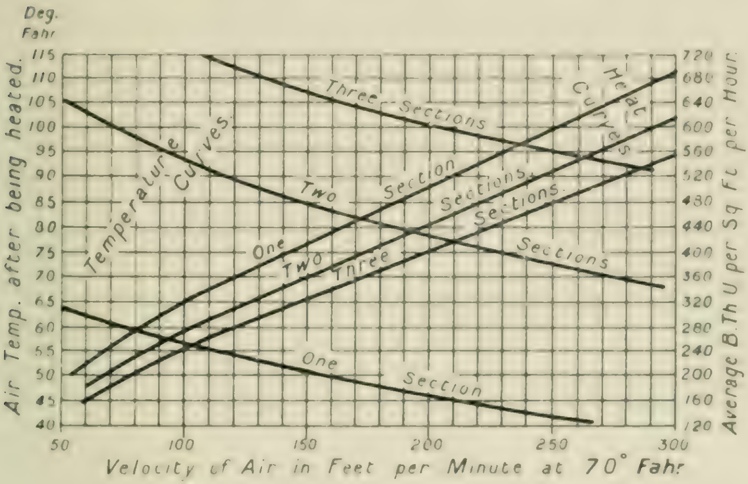


CHART 8.

Temperature to which air is warmed by "Regular Vento Heaters." Also heat emitted per square foot of surface per hour. Average water temperature, 180° Fahr. Temperature of entering air, 0° Fahr. Values taken from "Engineering Data on Vento Heaters."



When an ordinary radiator is located behind a wainscoting or similar covering, the heat emitted by it may be taken as from 50 to 75 per cent. of that given in Table XII., the exact value depending upon the provision made for the circulation of the air.

CHAPTER XIX

AREA OF HEATING SURFACE TO WARM BUILDINGS

DIFFERENT methods are adopted for ascertaining the amount of heating surface to produce a given effect, but in the more accurate ones, the total heat losses are estimated as closely as possible, whilst the latter values are divided by the heat transmitted per square foot of surface. This quantity may be expressed as a formula, or the various factors may be arranged in a tabular form.

Formula for estimating Direct Heating Surfaces.—

$$R = f\frac{(0{\cdot}016Q + Gk_1 + Wk_2 + Ak_3)(T - t)}{K} \quad . \quad . \quad (36)$$

where R = total heating surface in square feet.
f = variable factor for exposure, etc., Table IX.
Q = volume of air in cubic feet entering room per hour.
G = area of exposed glass in square feet.
W = area of exposed wall in square feet.
A = area of other important cooling surfaces in square feet.
T = internal temperature.
t = external temperature.
K = heat emitted per square foot of surface per hour.
$k_1\ k_2\ k_3$ = heat transmission coefficients agreeing with G, W, and A (see Chapter XVII.).

Where, however, definite values are assigned, as is usually the case in approximations, formula 36 may be simplified. For example, assume that the air exchange of a room and the external walls and glass form the principal cooling agencies, and the transmission coefficients for the two latter are taken

as 0·3 and 1·0 respectively. Let f = 1·1 and K = 150 for hot water heating and 250 for low-pressure steam heating.

For Hot Water Heating, when the internal temperature is 60° F. and that outside 30° —

$$R = \frac{Q}{300} + \frac{G}{4 \cdot 5} + \frac{W}{15} \quad . \quad . \quad . \quad . \quad (37)$$

70° inside and 0° outside—

$$R = \frac{Q}{120} + \frac{G}{2} + \frac{W}{6 \cdot 5} \quad . \quad . \quad . \quad . \quad (38)$$

For Low-Pressure Steam Heating.—Internal temperature 60° and external air 30° F.—

$$R = \frac{Q}{470} + \frac{G}{7 \cdot 6} + \frac{W}{25} \quad . \quad . \quad . \quad . \quad (39)$$

Internal temperature 70° and external air zero—

$$R = \frac{Q}{200} + \frac{G}{3 \cdot 2} + \frac{W}{11} \quad . \quad . \quad . \quad . \quad (40)$$

Formulæ 37 to 40 are, of course, only applicable for the conditions given. It will be found that the formulæ by different writers give different amounts of heating surface to produce the same effect, but there is nothing singular in this, as the difference in the coefficients used and the temperature range adopted may easily account for any seeming discrepancy. Moreover, the kind of surface used and its position affect the amount required. Formulæ 37 and 38 are based upon each square foot of surface transmitting 150 B.Th.U. per hour, and according to column 3 of Table XII., this coincides with that emitted from a two-column radiator for a temperature difference of 105°. In Great Britain, however, the external temperature is only as low as 30° F. for very brief periods, the average for the six coldest months being about 40° F. Now, if the internal temperature is maintained at 60°, the average amount of heat lost from an apartment is only about two-thirds of that allowed in formula 36. In other words, the average amount of heat transmitted per square foot of surface would only require to be 100 B.Th.U., which are emitted, according to Table XII., by

a two-column radiator with a temperature difference of about 75°. Thus the heating medium for average winter conditions would not require to exceed 75 + 60 = 135°, whilst for an external temperature of 30° F. it would require to be about 105 + 60 = 165° F.

When applying the general formula 36, it is not always desirable to take the lowest temperature recorded, for the thermal capacity of the apparatus and that of the building itself may be more than sufficient to meet any abnormally low temperature of short duration. As stated elsewhere, there is much uncertainty as to the volume of air that enters an apartment, and especially is this the case where vent flues are provided. Judgment, however, is the dominant factor in this matter, but where no special provision is made for ventilation, the air exchange per hour is often assumed as being from 1 to $1\frac{1}{2}$ times the capacity.

Example 20.—Determine the area of single-column radiator surface for warming a room with northern exposure when the remaining factors are as follows: Capacity, 4500 cubic feet; height, 15 feet; 80 square feet plate glass; 420 square feet of 14-inch brick wall; air exchanged twice per hour; external temperature 30° and that inside 60° F., whilst the average water temperature is 180° F.

By formula 36—

$$R = f\frac{(0{\cdot}016Q + Gk_1 + Wk_2)(T - t)}{K}.$$

From Table VIII., the glass coefficient is 0·75, that for the wall from Table VI. about 0·26, whilst the value of f from Table IX., may be assumed as 1·25. The mean difference between the heating medium and the temperature of apartment will be 180 − 60 = 120°, and for this Table XII. gives the emission per square foot of single-column radiator as 200 B.Th.U.

Substituting values,

$$R = 1{\cdot}25 \times \frac{\{(0{\cdot}016 \times 2 \times 4500) + (80 \times 0{\cdot}75) + (420 \times 0{\cdot}26)\} \times (60 - 30)}{200}$$

$$R = \frac{1{\cdot}25 \times 313 \times 30}{200},$$

when R = **59 square feet.**

Building.					Cooling agencies.								Temperature.			Heat lost.			Heating surfaces.					
Item.	Class of room.	Dimensions.	Aspect.	Capacity in cubic feet ×	Air exchange in terms of room capacity ×	Air coefficient =	Heat absorbed by air in B.Th.U. per deg. diff. of temp.	Kind of cooling surface.	Thickness of wall in ins.	Area of cooling surface in sq. ft. ×	Transmission coefficient =	Heat absorbed per deg. diff. of temp. in B.Th.U. ×	Temperature difference.	Internal temperature.	External temperature.	Heat absorbed per deg. diff. of temperature × temp. diff. ×	Factor f =	Total loss in B.Th.U.	Kind of surface.	Temperature of heating medium.	Temp. diff. of heating medium and apartment.	Heat emitted per sq. ft. of surface.	Total heating surface in sq. ft.	Remarks.
1	—	—	N.	4500	2	·016	144	Plate glass	—	80	·75	60	30	60	30				Single column 36" high					
								Wall	14	420	·26	109												
								For air exchange				144												
												313				9,390	1·25	11,740		180	120	200	59	
2	—	—	N.	4500	1	·016	72	Plate glass	—	80	·75	60	70	70	0				Single column 36" high					
								Wall	14	420	·26	109												
								For air exchange				72												
												241				16,870	1·25	21,000		180	110	175	120	

Example 21.—Consider the same room as in the previous example, but assume the internal temperature to be 70° and the external air to be zero, with one exchange of air per hour, the other values as before.

Here the temperature difference between the heating medium and the room is 180 − 70 = 110° F., and in Table XII. a single-column radiator for this difference emits 175 B.Th.U. per square foot per hour.

By formula 36—

$$R = f\frac{(0 \cdot 016Q + Gk_1 + Wk_2)(T - t)}{K}.$$

Substituting values,

$$R = 1 \cdot 25 \times \frac{\{(0 \cdot 016 \times 4500) + (80 \times 0 \cdot 75) + (420 \times 0 \cdot 26)\} \times 70}{175}$$

$$R = \frac{1 \cdot 25 \times 241 \times 70}{175};$$

when R = **120 square feet.**

The particulars for estimating the heating surface may be tabulated in any form that is convenient. A method is given on p. 219, and the values from Examples 20 and 21 are included.

Size of Ducts for Gravity Indirect Heaters.—It is desirable to regulate the size of these ducts for a minimum velocity which will vary according to their height. For this purpose, the following Table is given:—

TABLE XIII.

APPROXIMATE VELOCITIES THROUGH DUCTS OF VARYING HEIGHTS.

Height of duct in feet.	Velocity in feet per minute.	
	Mild climates.	Colder climates.
10	180	200
20	240	280
30	300	360
40	360	420
50	420	480
1	2	3

When the permissible velocity is known, the area of duct may be determined by the rule—

$$a = \frac{2 \cdot 4Q}{V} \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

where a = area of duct in square inches.

Q = cubic feet of air flowing through duct per hour.

V = air velocity in feet per minute.

The width of a duct, where possible, should be kept within twice its depth.

Size of Gravity Indirect Heaters.—The dimensions of an indirect heater are governed by its free area and that of the air velocity through it. The velocity of air through a gravity heater should be kept fairly low as a rule, or there may be difficulty in raising the air to the temperature desired. For giving the approximate size of Indirect Heater, formula 42 may be used, after which the correct dimensions can be readily obtained.

$$A = \frac{2 \cdot 4Q}{pV_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

where A = face area of heater in inches.

Q = volume of air in cubic feet flowing through heater per hour.

p = percentage area of free air space in terms of total face area.

V_1 = velocity of air through heater per minute.

Example 22.—Assuming the velocity through a heater to be 100 feet per minute and the volume of air delivered 15,000 cubic feet per hour, what size of heater section would be necessary when formed of 1-inch piping, the free air space being 53 per cent. of the face area?

By formula 42

$$A = \frac{2 \cdot 4Q}{pV_1}$$

Substituting values,

$$A = \frac{2 \cdot 4 \times 15{,}000}{0 \cdot 53 \times 100}$$

when A = **679 square inches** as the approximate face area.

Assuming now that the length of heater is fixed as 30 inches, then $\frac{679}{30} = 22{\cdot}66$ in. as the width. The latter value may require correcting, as the tubes would be a definite distance apart. For example, to give 53 per cent. of free area at the face of heater, tubes of 1 inch diameter would require to be spaced with $2\frac{3}{4}$-inch centres, whilst the pipes, if staggered as in Fig. 142, but with an equal number of tubes in each row, would take up nearly an additional $2\frac{3}{4}$ inches of width.

The number of tubes, therefore, in each row of a section for the case in hand will be obtained by dividing the approximate width by the spacing, which in round figures is $\frac{22{\cdot}66}{2{\cdot}75} = 8$. Correcting for the exact width, $2{\cdot}75 \times 8 = 22$ when the size of heater required in inches is 30 by 22.

To find the area of the heating surface in feet of 1 in. diameter iron tubing, divide the total length of piping by 3.

The calculations for indirect heaters are simplified by using standard sizes, as the following example shows:—

Example 23.—Take the same air volume and velocity as in the last example, but assume that a "Vento" indirect heater is used which has 44 per cent. of free air space at its face.

Applying formula 41—

$$a = \frac{2{\cdot}4Q}{V}.$$

Substituting values,

$$a = \frac{2{\cdot}4 \times 15{,}000}{100},$$

when $a = $ **360 square inches or 2·5 square feet**, as the required free air space at the face of section.

Each "regular 40-inch" loop is listed as having a length of $41\frac{1}{16}$ inches, and containing at its face 0·62 square foot of free air space, the loops being 5 inches in width. The number of loops to give the free air space required will be $\frac{2{\cdot}5}{0{\cdot}62} = 4{\cdot}03$, or say 4, and the width occupied by the loops when joined will be $5 \times 4 = 20$ inches, giving the total face size of section in inches as $41\frac{1}{16}$ by 20.

Application of Charts for Indirect Heaters.—To do this, two further examples will be considered in which the air volume to produce a given effect and the size of the ducts are both taken in account.

Example 24.—Two adjoining rooms with northern exposure and 11 feet in height are warmed with gravity indirect heaters. *Room No.* 1 has 120 square feet of 18-inch brick wall, 70 square feet sheet glass, whilst the inward leakage of cold air will be assumed as 800 cubic feet per hour. *Room No.* 2 has 220 square feet of 18-inch brick wall, 130 square feet of sheet glass, and an assumed inward air leakage of 1200 cubic feet per hour. The velocities of air through ducts and heaters to be taken as 180 and 90 feet per minute respectively. Internal temperature to be maintained at 60° Fahr., when that outside is 30° Fahr. Air temperature at registers for these conditions to be about 90° Fahr., the heater being formed in sections of 1-inch piping, spaced with $2\frac{3}{4}$-inch centres. Steam pressure in heater 5 lb. per square inch, and the free area of the registers 60 per cent. of their total surface.

Determine—(*a*) Heat lost by each room per hour.
(*b*) Volume of air to maintain temperature of each room.
(*c*) Size of air ducts.
(*d*) Size of registers.
(*e*) Size of intake duct.
(*f*) Size of heater.

The heat lost may be determined by formula 32,

$$\text{where } U = f(0 \cdot 016Q + Gk_1 + Wk_2 + Ak_3)(T - t).$$

For this f may be taken from Table IX. as 1·15.

Substituting values for Room No. 1,

$$U = 1 \cdot 15\{(0 \cdot 016 \times 800) + (70 \times 1) + (120 \times 0 \cdot 23)\} \times (60 - 30),$$
$$U = 1 \cdot 15 \times 110 \times 30,$$
when U = **3800 B.Th.U. per hour nearly.**

Substituting values for Room No. 2,

$$U = 1 \cdot 15\{(0 \cdot 016 \times 1200) + (130 \times 1) + (220 \times 0 \cdot 23)\} \times (60 - 30),$$
when U = **6900 B.Th.U. per hour.**

Volume of Air to maintain Temperature is found by formula 26,

$$\text{where } Q = \frac{56U}{T_a - T}$$

Substituting values for Room No. 1,

$$Q = \frac{56 \times 3800}{90 - 60},$$

when Q = **7100 cubic feet per hour.**

Substituting values for Room No. 2,

$$Q = \frac{56 \times 6900}{90 - 60},$$

when Q = **12,880 cubic feet per hour.**

Size of Air Ducts to Apartments may be determined by formula 41,

$$\text{where } a = \frac{2{\cdot}4Q}{V}$$

Substituting values for Room No. 1,

$$a = \frac{2{\cdot}4 \times 7100}{180},$$

when a = **95 square inches**, or 12″ × 8″ opening.

Substituting values for Room No. 2,

$$a = \frac{2{\cdot}4 \times 12{,}880}{180},$$

when a = **172 square inches**, or 12″ × 15″ opening.

Size of Registers is found by dividing cross-sectional area of ducts by 0·6.

For Room No. 1,

Area of register $= \frac{95}{0{\cdot}6} =$ **158 square inches**, say size 9″ × 18″.

For Room No. 2,

Area of register $= \frac{172}{0{\cdot}6} =$ **286 square inches**, say size 14″ × 22″.

Size of Intake Duct.—Assuming the air velocity as before, the area of this will be equal to the sum of the areas of the "up-cast" ducts, which for the two under consideration will be $95 + 172 =$ **267 square inches.**

Size of Heater.—The approximate face area of heater may be found by formula 42,

$$\text{where } A = \frac{2 \cdot 4Q}{pV_1}$$

The velocity through the heater is given as 90 feet per minute, and the value of p when the tubes are spaced with $2\frac{3}{4}$-inch centres is about 0·53. The volume of air for both rooms is $7100 + 12{,}880 = 19{,}980$ cubic feet per hour.

Substituting values,

$$A = \frac{2 \cdot 4 \times 19{,}980}{0 \cdot 53 \times 90},$$

when $A =$ **1006 square inches.**

Assuming the heater to take a form similar to that in Fig. 142 with tubes 40 inches in length, its approximate width will be $\frac{1006}{40} = 25 \cdot 1$ inches. The latter value divided by 2·75 gives 9·1, or 9 tubes per row, whilst the exact width will be $2 \cdot 75 \times 9 = 24 \cdot 75$ inches. The face of a section in inches will therefore be 40 by $24\frac{3}{4}$.

The number of sections is found from chart 1, p. 211, which, for a velocity of 90 feet per minute, gives two. The final temperature, however, that may be obtained by two sections with air entering at 30° Fahr. is given as 100° Fahr., so that the supply of steam will need regulation. This may be readily done by the arrangement shown in Fig. 142, and for milder weather, one section may be cut out of use.

Example 25.—Assume now that the two rooms in Example 24 require to be maintained at 70° Fahr. when the external air is zero. The air velocities through the ducts and heater to be taken as 200 and 100 feet per minute respectively. Let the heater be the "Vento" type, as illustrated in Fig. 143, whilst the remaining particulars are as before.

Heat lost from Rooms.—

By formula 32,

$$U = f(0 \cdot 016Q + Gk_1 + Wk_2 + A'k_3)(T - t)$$

Q

Substituting values for Room No. 1,

$$U = 1{\cdot}15\{(0{\cdot}016 \times 800) + (70 \times 1) + (120 \times 0{\cdot}23)\} \times (70 - 0)$$

$$U = 1{\cdot}15 \times 110 \times 70,$$

when $U =$ **8860 B.Th.U. per hour.**

Substituting values for Room No. 2,

$$U = 1{\cdot}15\{(0{\cdot}016 \times 1200) + (130 \times 1) + (220 \times 0{\cdot}23)\} \times (70 - 0)$$

$$U = 1{\cdot}15 \times 200 \times 70,$$

when $U =$ **16,100 B.Th.U. per hour.**

Volume of Air required to maintain Temperature of Rooms.—By formula 26,

$$Q = \frac{56U}{T_a - T}$$

Substituting values for Room No. 1,

$$Q = \frac{56 \times 8860}{120 - 70},$$

when $Q =$ **9920 cubic feet per hour.**

Substituting values for Room No. 2,

$$Q = \frac{56 \times 16{,}100}{120 - 70},$$

when $Q =$ **18,000 cubic feet per hour.**

Sizes of Ducts to Registers.—By formula 41,

$$a = \frac{2{\cdot}4Q}{V}.$$

Substituting values for Room No. 1,

$$a = \frac{2{\cdot}4 \times 9920}{200},$$

when $a =$ **119 square inches**, say 14″ × 9″.

Substituting values for Room No. 2,

$$a = \frac{2{\cdot}4 \times 18{,}000}{200},$$

when $a =$ **216 square inches**, say 18″ × 12″.

Size of Registers.

For Room No. 1,

Area of register $= \dfrac{119}{0{\cdot}6} =$ **198 square inches**, say size $10'' \times 20''$.

For Room No. 2,

Area of register $= \dfrac{216}{0{\cdot}6} =$ **360 square inches**, say, size $16'' \times 24''$.

Size of Fresh Air Duct.

Free area $= 119 + 216 =$ **335 square inches.**

Size of Heater.—The air velocity through heater being half that in the ducts, the free area of heater should be double that of the fresh air duct, or

$335 \times 2 =$ **670 square inches** $=$ **4·65 square feet.**

Each 40-inch "Regular Vento" loop contains 0·62 square foot of free air space, therefore the number of loops to give the area desired $= \dfrac{4{\cdot}65}{0{\cdot}62} =$ **7·5 or 8.** For these loops, the total width of a section will be $8 \times 5 = 40$ inches.

By reference to chart 7, p. 214, two sections will be required to raise the air from 0° to 120° Fahr., with steam at 5 lb. gauge pressure, when the velocity is 100 feet per minute.

The values obtained for Examples 24 and 25 may be tabulated thus—

Room No.	Internal temp.	External temp.	Temp. of air at registers.	Total heat lost per hour in B.Th.U.	Volume of air in cubic feet delivered by registers.	Area of duct in inches.	Size of registers in inches.
1	60	30	90	3,800	7,100	95	9 × 18
2	60	30	90	6,900	12,880	172	14 × 22
1	70	0	120	8,860	9,920	119	10 × 20
2	70	0	120	16,100	18,000	216	16 × 24

Room No.	Area of fresh air duct in inches.	Type of heater	Size of heaters.	Free area of heater in inches.	Number of sections
1, 2	267	Tubular	40″ × 24[illegible]	525	2
1, 2	335	"Vento"	$41\frac{1}{64}''$ × 40	722	2

CHAPTER XX

SIZING PIPES FOR GRAVITY SYSTEMS OF HOT-WATER HEATING

In designing an installation, the piping should be sized, so that the total resistance is duly proportioned between the different circuits. If this is attained, one circuit will work just as well as another, even when the smallest permissible sizes of pipes are used.

By the aid of formulæ, and a keen insight into the work in hand, the resistances of the different circuits may be proportioned in a manner that is not possible by other means. The personal factor, however, which cannot be expressed by an equation, is a very important one when applying formulæ to the various piping systems.

The writer's general formula is expressed as follows:—

$$U = w_3 t_1 c_1 \sqrt{\frac{d^5 h\left(\frac{w_2}{w_1} - 1\right)}{l}} \quad . \quad . \quad . \quad . \quad (43)$$

from which

$$d = \sqrt[5]{\frac{l}{h\left(\frac{w_2}{w_1} - 1\right)} \times \left(\frac{U}{w_3 t_1 c_1}\right)^2} \quad . \quad . \quad . \quad (44)$$

where U = British thermal units per hour.

w_1 = weight per cubic foot of water at the average temperature of that in flow pipe.

w_2 = weight per cubic foot of water at the average temperature of that in return pipe.

w_3 = weight per cubic foot at the mean temperature of the flow and return waters.

where d = diameter of pipe in inches.
h = effective circuit height in feet.
l = length of circuit in feet.
t_1 = temperature difference of water between leaving and re-entering boiler or other heater.
c_1 = a coefficient which varies with the size of piping.

TABLE XIV.

VALUE OF c_1.

Diameter of pipe. Inches.	Value of c_1.	Diameter of pipe Inches.	Value of c_1.
$\frac{1}{4}$	175	$2\frac{1}{2}$	232
$\frac{3}{8}$	178	3	240
$\frac{1}{2}$	184	$3\frac{1}{2}$	246
$\frac{3}{4}$	192	4	255
1	198	$4\frac{1}{2}$	260
$1\frac{1}{4}$	206	5	263
$1\frac{1}{2}$	212	6	272
2	224	8	284

Example 26.—What is the capacity in B.Th.U. of a "One Pipe" circuit 4 inches diameter and 300 feet long, where the effective circuit height is 15 feet, the difference of temperature between leaving and re-entering boiler 40° F., average temperature of flow water 170° F., and that of the return 150° F.?

By formula 43

$$U = w_3 t_1 c_1 \sqrt{\frac{d^5 h\left(\frac{w_2}{w_1} - 1\right)}{l}}$$

Mean temperature of flow and return water equals $\frac{170 + 150}{2} = 160°$ F. The values coinciding with w_1, w_2, and w_3 will be found to be 60·78 lb., 61·2 lb., and 60·99 lb. respectively, whilst the value of c_1 for a 4-inch circuit is given as 255.

Substituting values—

$$U = 60{\cdot}99 \times 40 \times 255 \sqrt{\frac{4^5 \times 15 \times \left(\frac{61{\cdot}2}{60{\cdot}78} - 1\right)}{300}}$$

$$U = 60{\cdot}99 \times 40 \times 255 \times 0{\cdot}594;$$

when $U =$ **369,800 B.Th.U.**

For general work, formulæ 43 and 44 are rather cumbersome, but they may be readily simplified when a certain range of conditions is decided upon. For example, let the formulæ be required for a drop of temperature of 40° between the water leaving and re-entering heater, when the average temperature of the "flow" is 165° F. and that of the return 135° F.

Formula for a Temperature Drop of 40° *F.*

$$U = 248c_1\sqrt{\frac{d^5h}{l}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

$$\text{and } d = \sqrt[5]{\frac{l}{h} \times \left(\frac{U}{248c_1}\right)^2} \quad . \quad . \quad . \quad . \quad (46)$$

Resistance of Pipe Fittings.—So far, the only resistance that has been taken into account is that of pipe friction, but as bends and other fittings also impede the motion of water through pipes, these require to be taken into consideration as well.

To simplify the process of calculation, it is usual to express the resistance of any fitting or connection in terms of that offered by a sharp elbow, which is taken as the unit of resistance. In turn, this unit of resistance is often expressed in feet of pipe which offers an equivalent amount of retardation, and it is added to the circuit in question.

The resistance offered by a sharp elbow is usually obtained by the formula—

$$h_1 = \frac{v^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

where h_1 = head in feet absorbed by friction.
v = velocity in feet per second.
g = acceleration of gravity = 32·2.

If the head absorbed by an elbow is expressed in length of piping which offers the equivalent retardation, then—

$$l_1 = \left(\frac{c_1}{1.57}\right)^2 \times d \quad . \quad . \quad . \quad . \quad (48)$$

where l_1 = equivalent length of pipe in feet.

d = diameter of elbow in inches.

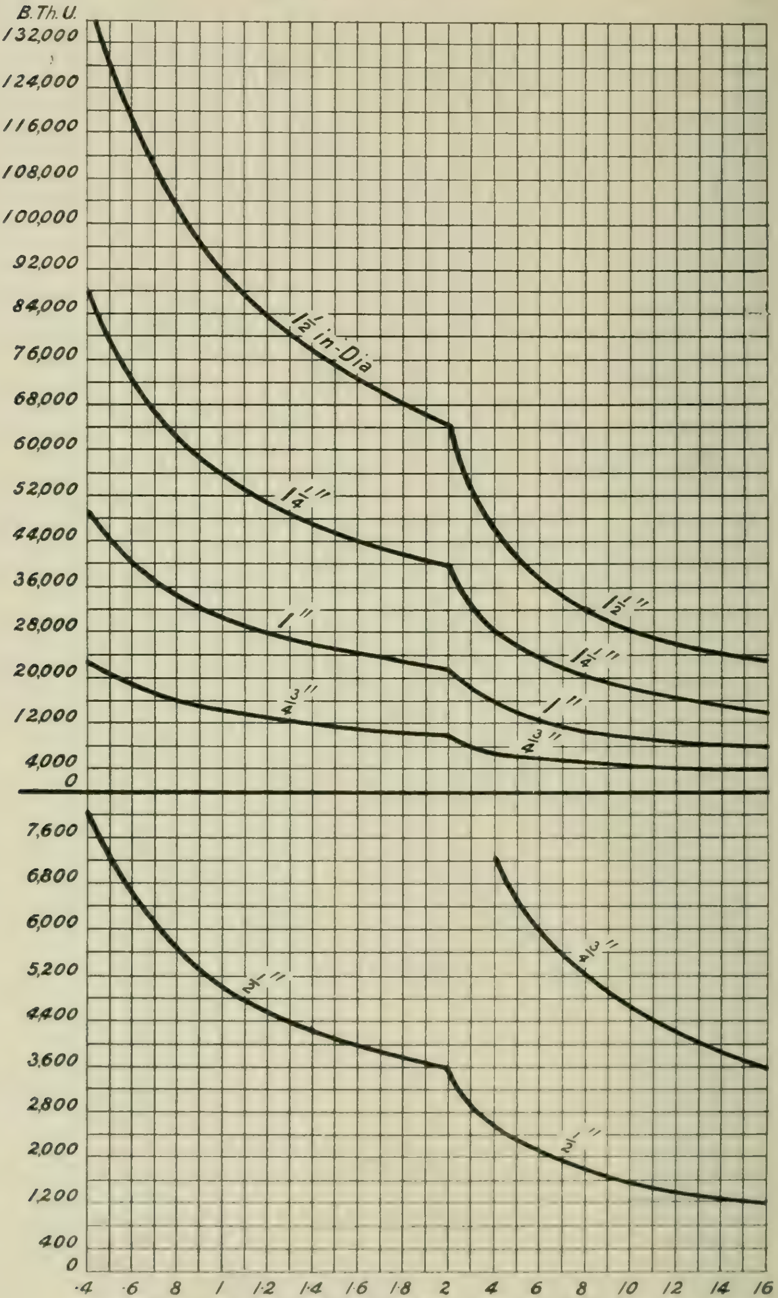
c_1 = value from Table XIV.

It is the opinion of the writer that the values obtained by formula 48 are not quite high enough for elbows of small bore, but a little too liberal for the larger sizes. Table XXIII. in the appendix has, therefore, the values adjusted, whilst the single resistances for the different fittings are in the main those adopted by Prof. Rietschell.

To facilitate the sizing of pipe, Charts 9 to 16 and Tables XV. and XVI. have been prepared. Charts 9 to 11 are for a temperature drop of 30° F., and Charts 12 to 14 for a fall of 40° F. These are chiefly intended for estimating the mains of "up-feed" systems, whilst the values should be increased by 10 per cent. for the mains and branches of "down-feed" installations. Charts 15 and 16 are for sizing risers of "up-feed" systems and for temperature differences of 25° and 35° F. respectively. By the use of the latter charts, the risers may be figured for a smaller temperature difference than the mains, a procedure that should be followed in general work. To comply strictly with actual cooling conditions, a greater temperature difference should be allowed for the nearer risers when systems take the forms indicated by Figs. 144 and 145. This, however, may be done by simple adjustment and without the use of further charts.

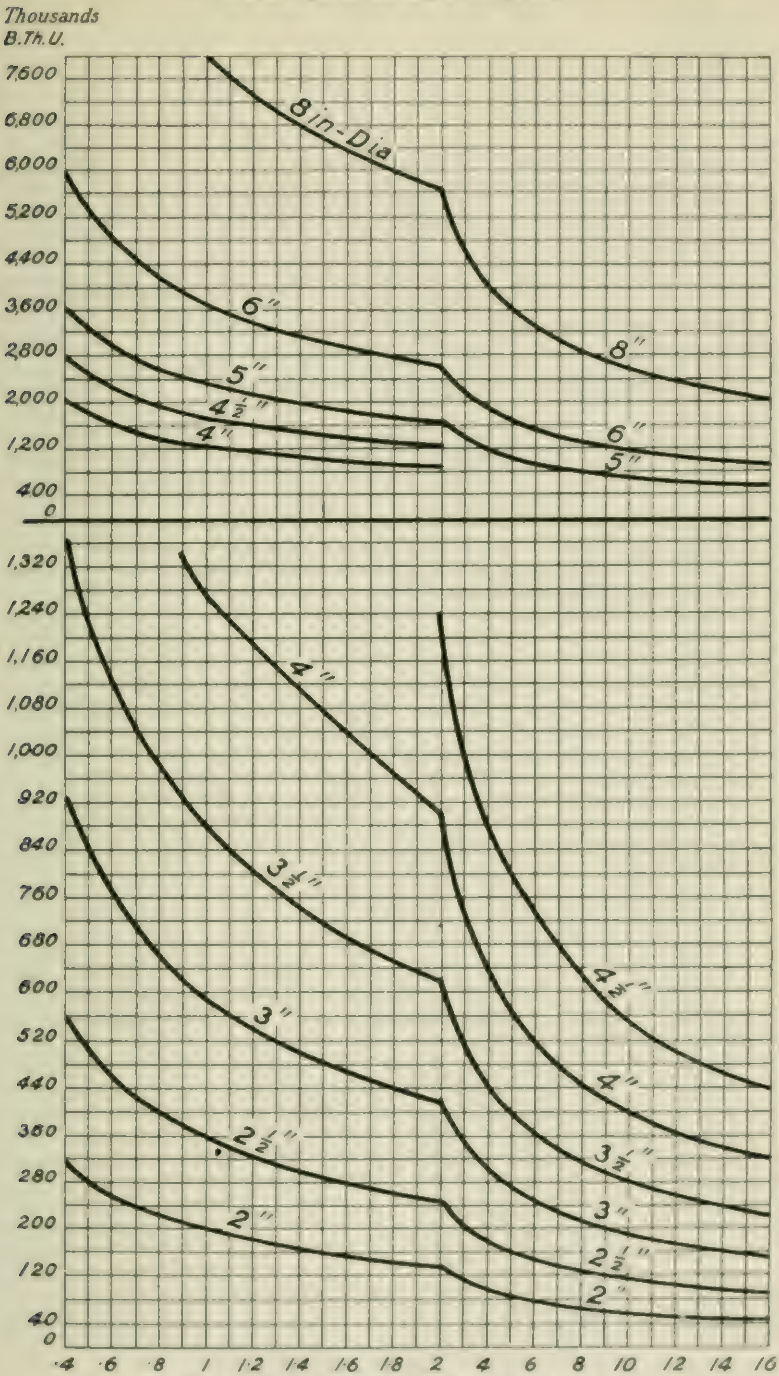
The tables are convenient for obtaining rapidly approximate values.

CHART 9.—Low-pressure hot water. Capacity of circuits in B.Th.U. per hour for a temperature drop of 30° Fahr.



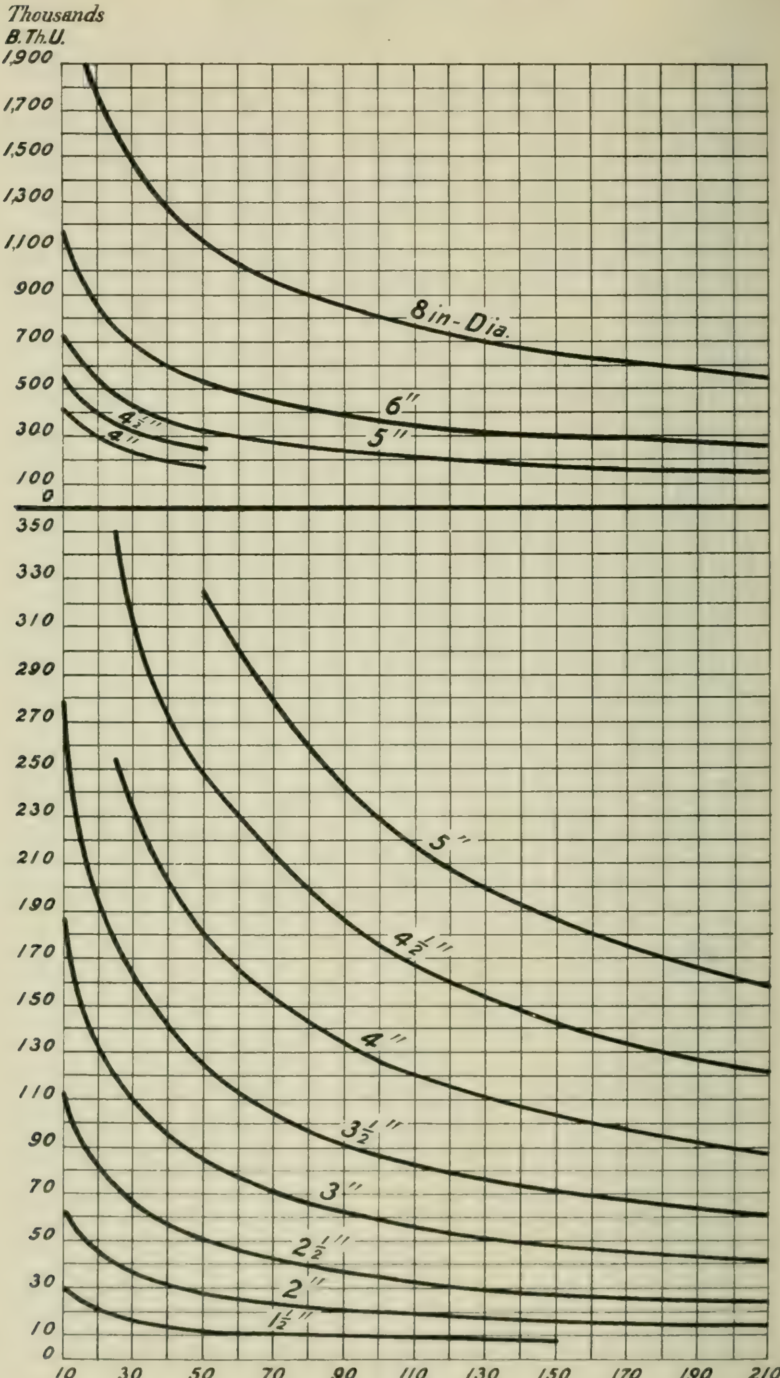
Ratios: Length of circuit ÷ effective height in feet.

Chart 10.—Low-pressure hot water. Capacity of circuits in B.Th.U. per hour for a temperature drop of 30° Fahr.



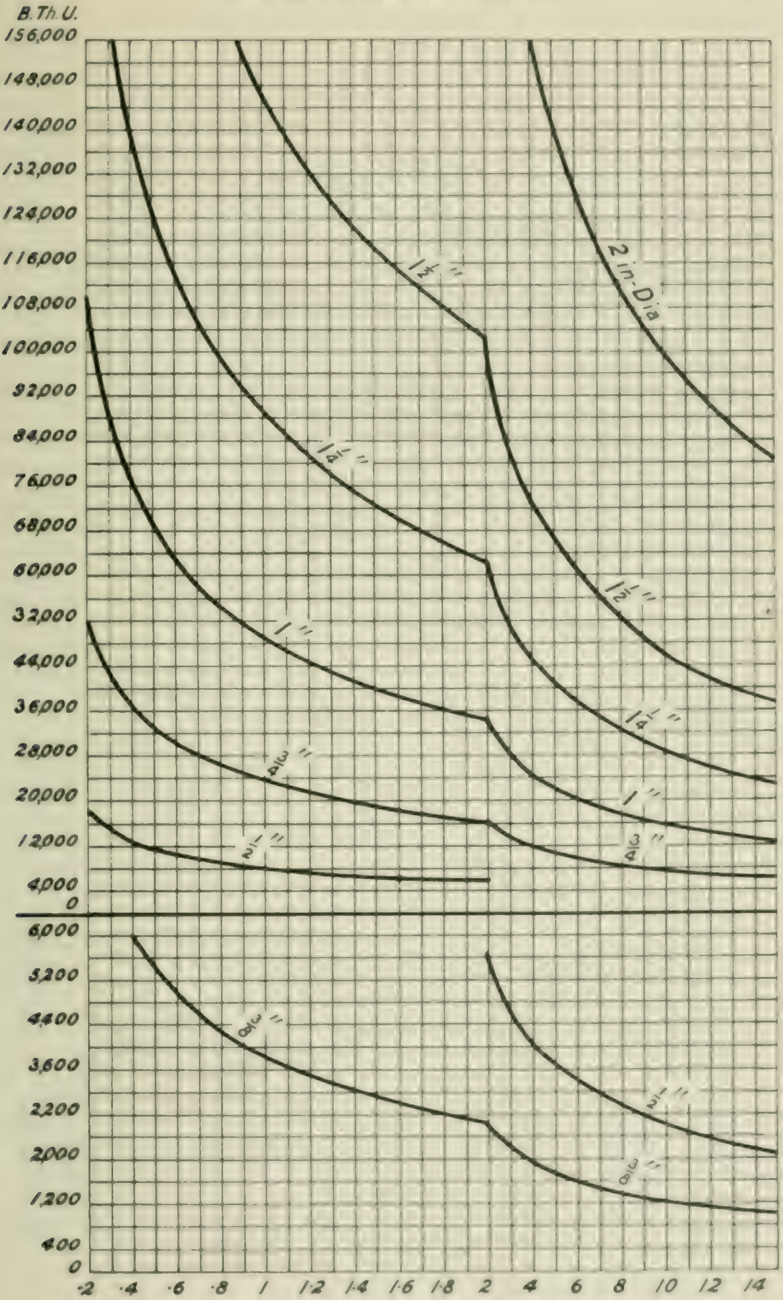
Ratios: Length of circuit ÷ effective height in feet.

CHART 11.—Low-pressure hot water. Capacity of circuits in B.Th.U. per hour for a temperature drop of 30° Fahr.



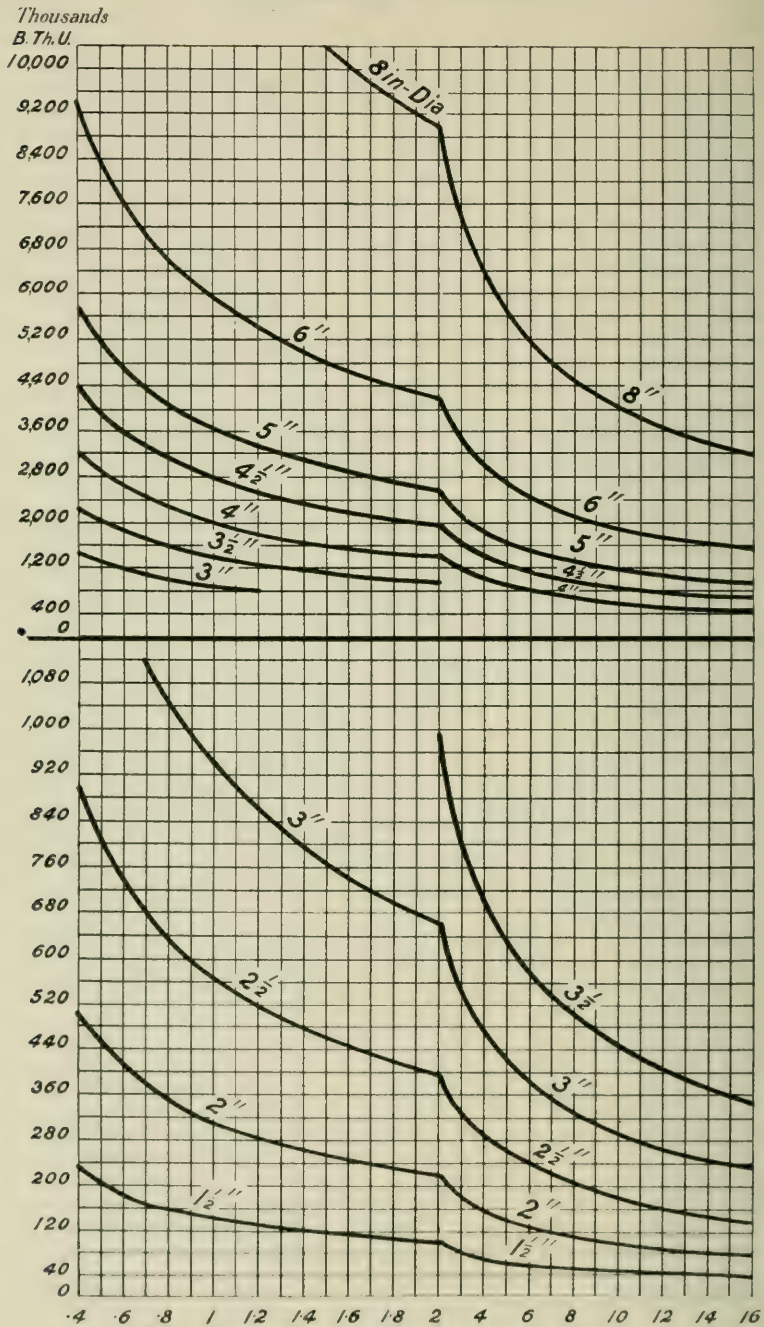
Ratios: Length of circuit ÷ effective height in feet.

Chart 12.—Low-pressure hot water. Capacity of circuits in B.Th.U. per hour for a temperature drop of 40° Fahr.



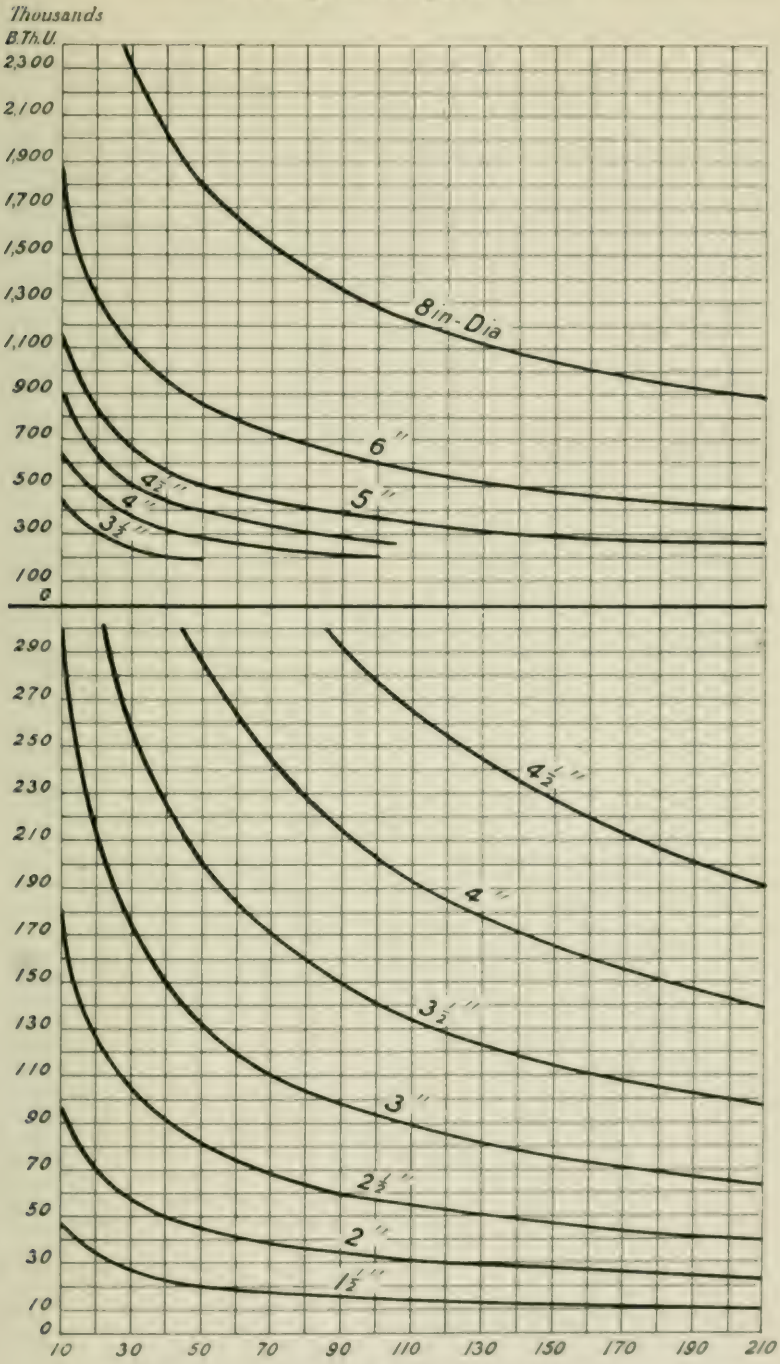
Ratios. Length of circuit ÷ effective height in feet.

CHART 13.—Low-pressure hot water. Capacity of circuits in B.Th.U. per hour for a temperature drop of 40° Fahr.



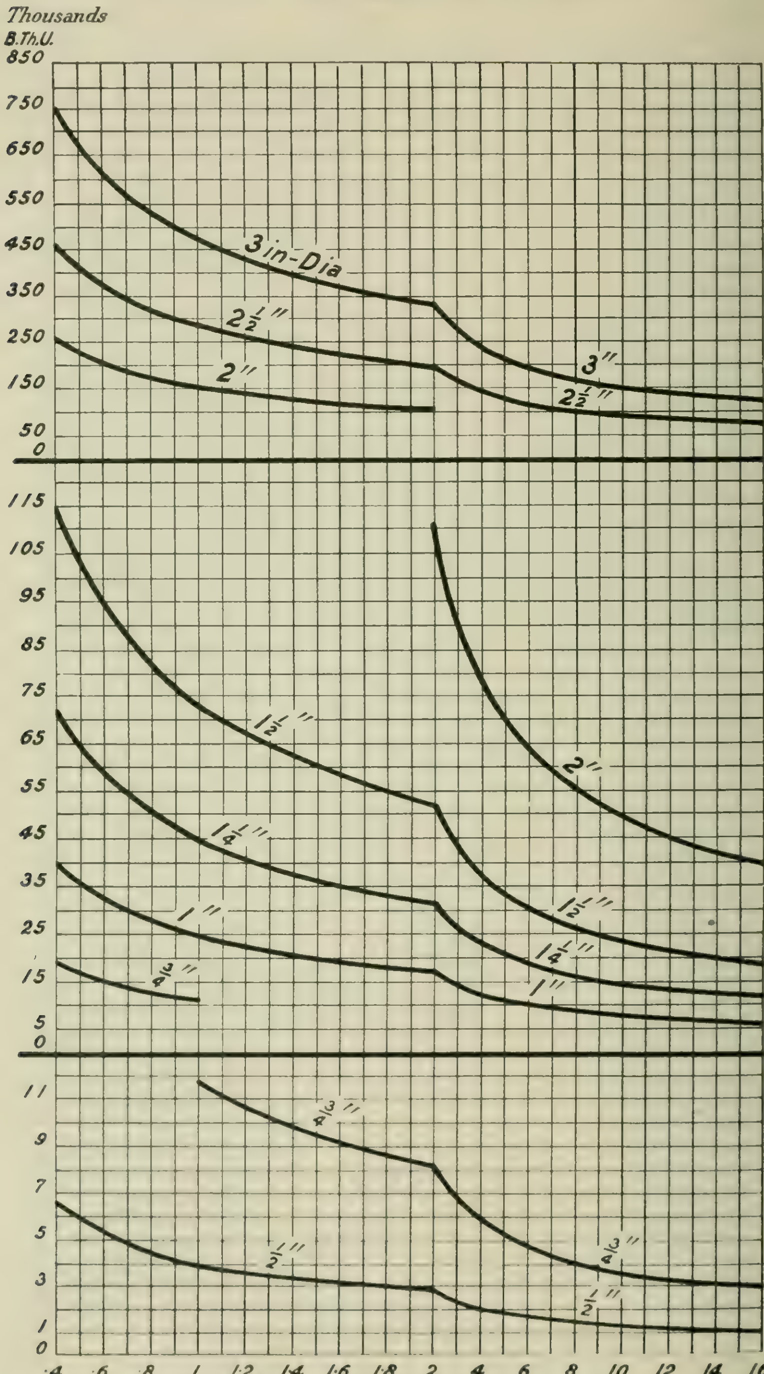
Ratios: Length of circuit ÷ effective height in feet.

CHART 14. Low-pressure hot water. Capacity of circuits in B.Th.U. per hour for a temperature drop of 40° Fahr.



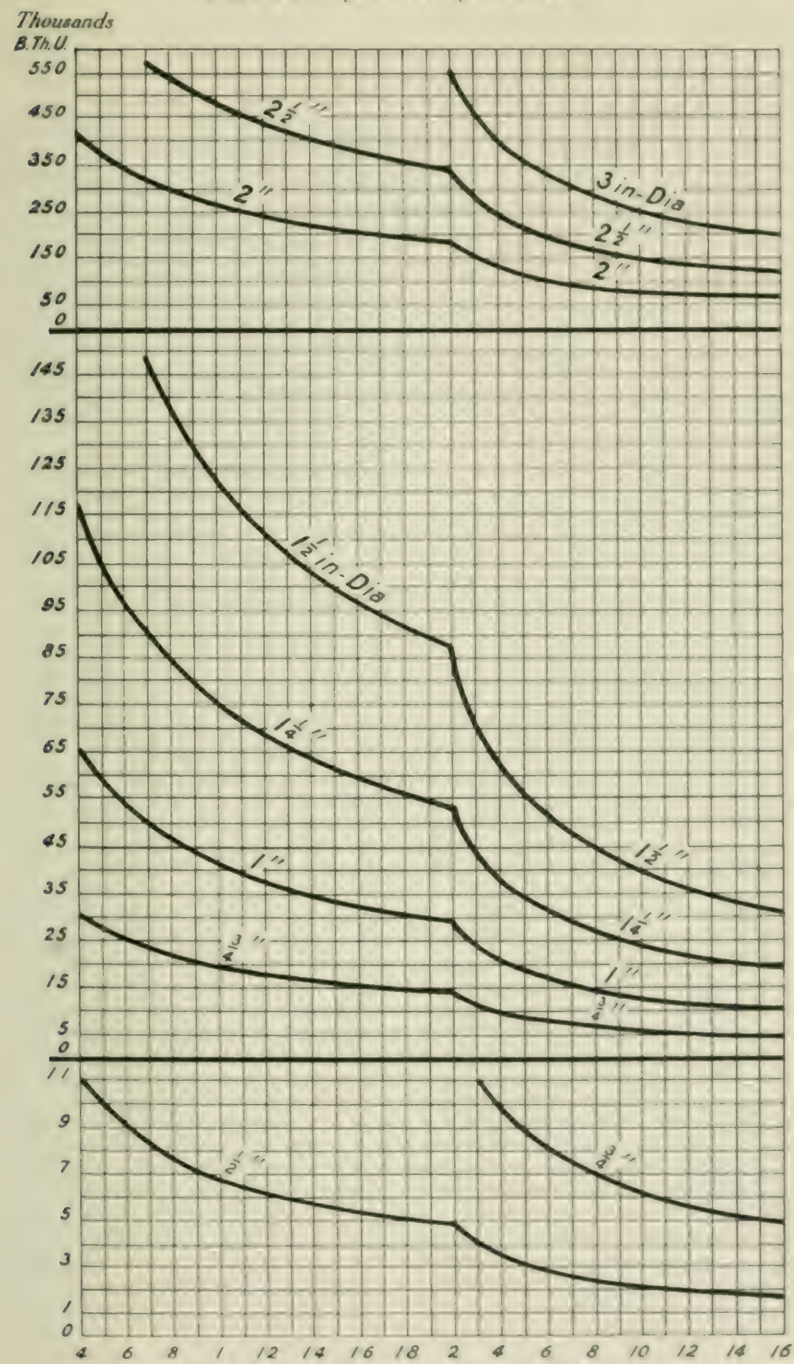
Ratios: Length of circuit ÷ effective height in feet.

CHART 15.—Low-pressure hot water. Capacity of riser circuits in B.Th.U. per hour for a temperature drop of 25° Fahr.



Ratios: Length of circuit ÷ effective height in feet.

CHART 16.—Low-pressure hot water. Capacity of riser circuits in B.Th.U. per hour for a temperature drop of 35° Fahr.



Ratios: Length of circuit ÷ effective height in feet.

TABLE XV.

Approximate Capacity of Main Circuits in British Thermal Units, and in Square Feet of Heating Surface for a circuit height of 10 feet, and for a temperature drop of 30° F. Each square foot of surface is assumed as transmitting 140 B.Th.U. per hour.

Diameter in inches.	Length of circuit in feet.										
	20	40	60	90	120	160	250	500	700	1,000	
1	21,900 156	15,500 110	12,700 91	10,300 73	8,950 64	7,750 55	6,200 44	— —	— —	— —	B.Th.U. Sq. ft.
1¼	39,600 280	28,000 200	22,900 163	18,700 133	16,200 116	14,000 100	11,200 80	7,930 56	— —	— —	B.Th.U. Sq. ft.
1½	65,000 465	46,000 330	37,500 268	30,700 220	26,600 190	23,000 164	18,400 132	13,000 93	11 000 78	— —	B.Th.U. Sq. ft.
2	140,000 1,000	99,000 710	81,000 580	66,000 470	57,200 410	49,500 354	39,600 284	28,000 200	24,700 175	19,800 140	B.Th.U. Sq. ft.
2½	253,000 1,800	179,000 1,280	146,000 1,040	119,000 850	103,000 735	89,500 640	71,600 510	50,700 358	42,700 305	35,800 256	B.Th.U. Sq. ft.
3	418,000 3,000	296,000 2,120	242,000 1,730	197,000 1,400	171,000 1,220	148,000 1,060	118,400 845	83,700 600	71,000 507	59,200 420	B.Th.U. Sq. ft.
3½	622,000 4,450	440,000 3,140	360,000 2,570	293,000 2,090	254,000 1,820	220,000 1,570	176,000 1,260	124,000 885	105,000 755	88,000 630	B.Th.U. Sq. ft.
4	899,000 6,400	635,000 4,530	518,000 3,700	423,000 3,020	367,000 2,620	317,000 2,260	254,000 1,810	180,000 1,280	152,000 1,080	127,000 910	B.Th.U. Sq. ft.
4½	1,240,000 8,850	875,000 6,250	715,000 5,100	583,000 4,150	505,000 3,600	437,000 3,120	350,000 2,500	248,000 1,770	209,000 1,490	175,000 1,250	B. Th.U. Sq. ft.
5	1,620,000 11,500	1,150,000 8,200	938,000 6,700	766,000 5,460	665,000 4,750	575,000 4,100	460,000 3,280	325,000 2,300	275,000 1,960	230,000 1,640	B.Th.U. Sq. ft.
6	2,640,000 18,800	1,870,000 13,300	1,520,000 10,800	1,250,000 8,950	1,080,000 7,700	935,000 6,660	748,000 5,350	530,000 3,780	445,000 3,180	374,000 2,670	B.Th.U. Sq. ft.
8	5,680,000 40,500	4,020,000 28,700	3,280,000 23,400	2,680,000 19,100	2,320,000 16,600	2,010,000 14,300	1,608,000 11,400	1,140,000 8,150	962,000 6,860	804,000 5,730	B.Th.U. Sq. ft.
1	2	3	4	5	6	7	8	9	10	11	12

TABLE XVI.

APPROXIMATE CAPACITY OF COMPOUND RISERS IN BRITISH THERMAL UNITS, AND IN SQUARE FEET OF HEATING SURFACE FOR A TEMPERATURE DROP OF 25° F. EACH SQUARE FOOT OF SURFACE IS ASSUMED AS TRANSMITTING 140 B.TH.U. PER HOUR.

Diameter in inches.	Height of circuit in feet.									
	4	10	20	30	40	50	60	70	80	
½	—	—	—	—	4,060	4,570	4,850	5,240	5,740	B.Th.U.
					29	32	34	37	41	Sq. ft.
¾	3,370	5,230	8,300	10,350	11,700	13,100	14,000	15,100	16,500	B.Th.U.
	24	37	59	74	83	93	100	108	118	Sq. ft.
1	7,160	11,050	17,500	21,700	24,800	28,000	29,600	32,000	35,000	B.Th.U.
	51	79	125	155	177	200	212	229	250	Sq. ft.
1¼	13,000	20,000	31,800	39,400	45,000	50,300	53,800	58,000	63,600	B.Th.U.
	93	143	227	282	320	360	384	415	454	Sq. ft.
1½	21,200	32,800	52,000	64,500	73,500	82,200	87,800	95,000	104,000	B.Th.U.
	150	234	370	460	525	585	625	680	740	Sq. ft.
2	45,600	70,600	111,000	138,000	158,000	176,000	189,000	204,000	223,000	B.Th.U.
	326	503	790	990	1,130	1,250	1,350	1,460	1,590	Sq. ft.
2½	82,500	127,000	202,000	250,000	286,000	320,000	342,000	370,000	404,000	B.Th.U.
	590	910	1,450	1,780	2,040	2,280	2,440	2,640	2,880	Sq. ft.
3	137,000	212,000	335,000	417,000	475,000	530,000	567,000	614,000	672,000	B.Th.U.
	980	1,510	2,390	2,900	3,400	3,790	4,050	4,370	4,800	Sq. ft.
1	2	3	4	5	6	7	8	9	10	11

Equivalent Resistance of Pipes.—There are many different ways in which the sizing of pipes for heating systems is effected, but the one the writer has adopted consists of two processes. In the first, approximate sizes are rapidly figured, after which corrections are made by ascertaining the resistance of the piping in terms of any particular diameter. Table XXIV. in the appendix has been prepared to aid in this operation. Assume, for example, that a portion of a circuit consists of 50 feet 4-inch pipe, 40 feet 3-inch pipe, and 60 feet of 2-inch diameter pipe, and a given weight of water is circulated through the series. To estimate the capacity or resistance of such an arrangement in the ordinary way requires a rather long and involved process; if, however, only one size is presented, the case is a simple one, and admits of its capacity being directly obtained by means of one of the charts, or of being calculated by the formulæ given. Suppose now the circuit in question is expressed as the equivalent of 2-inch diameter piping. Consulting Table XXIV. of appendix, it will be observed that the proportional resistance of 3-inch diameter piping in terms of 2 in. diameter is given as 0·115 and that of 4-inch diameter piping as 0·024. Thus $(50 \times 0{\cdot}024) + (40 \times 0{\cdot}115) + 60 = 65{\cdot}8$, or say 66 feet of 2-inch diameter pipe offer the equivalent resistance to the three pipes in question. If the equivalent resistance is expressed in terms of 4-inch diameter piping, then the proportional retardation per foot for the 3-inch and the 2-inch diameter pipes is given as 4·74 and 41 feet respectively. Thus under the latter conditions, the equivalent resistance is offered by $50 + (40 \times 4{\cdot}74) + (60 \times 41) = 2700$ feet of 4-inch diameter pipe.

Method of Sizing "One-Pipe Up-Feed" Systems.—Let Fig. 144 be the installation to be sized for a temperature drop in the main circuits of 30° F., and for an average of 25° F. in the risers. The effective circuit height h for the mains is the vertical distance between the boiler fire grate and the highest point, say at the bend above A. For the risers, the effective circuit heights are indicated by h_1 and h_2, these being the vertical distances between the centres of the heating surfaces and the main piping.

The tabulated particulars of Fig. 144 indicate the procedure

that may be followed in determining the sizes of the pipes. At the outset, the circuit lengths are roughly computed or assumed and these are noted in column 4.

Take section AB. This supplies both the circuits shown, the longer of which is 250 feet. To this add 50 feet, say, for the proportional resistance of bends and boiler connections,

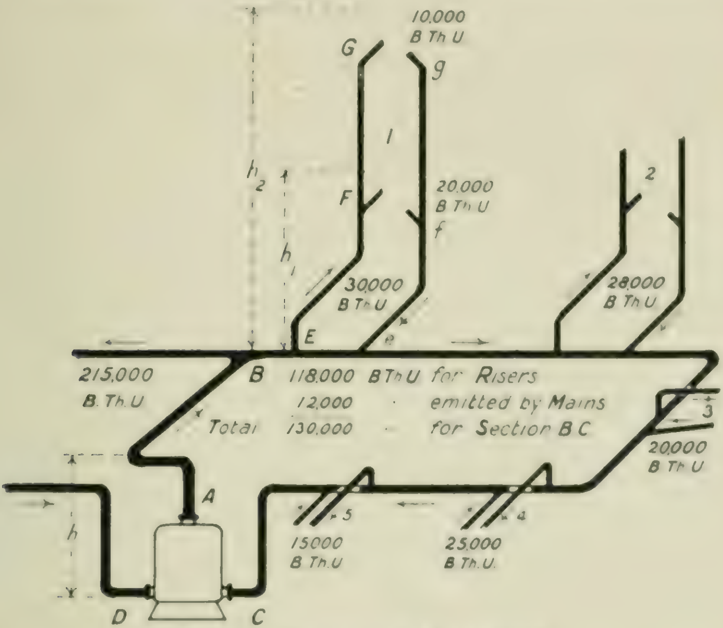


FIG. 144.—"One-pipe up-feed" system.

giving a total length of 300 feet. The effective circuit height is 10 feet, and AB must supply (215,000 + 130,000) = 345,000 B.Th.U. Dividing length by height, 30 are obtained, and from Chart 11 a 5-inch diameter pipe is selected, the size below being rather small unless a greater temperature drop is allowed.

Section BD is required to carry 215,000 B.Th.U., and for a ratio of 30 a 4-inch pipe will be necessary.

Section BC is called upon to supply 130,000 B.Th.U., and if to the length ABC 40 feet are added for bends, etc.,

190 feet are obtained. The ratio in this case is $\frac{190}{10} = 19$, and from Chart 11, it will be found that a 3-inch diameter pipe has the requisite capacity.

Columns 7 to 10 show how the total length for each section is estimated, but this method of checking the sizes is not so necessary for the mains of "one-pipe" systems as for risers and "two-pipe" systems.

Riser No. 1 (Fig. 144) may now be roughly sized. The length of EF*fc* is 30 feet, and the circuit height to F 4 feet. Assume a length of 50 feet so as to cover the resistance of fittings, etc.,

PARTICULARS OF FIG. 144.

Part of system.	Section.	Capacity in British Thermal Units.	Trial length of circuit l.	Approx. value of $\frac{l}{h}$.	Trial dia. in ins. d.	Net length of section. +	Equivalent length for fittings, etc. +	Proportional length of circuit in terms of section required =	Total length of circuit l.	Effective circuit height h.	Length of circuit divided by height $\frac{l}{h}$.	Corrected diameter in inches d.
Main	AB	345,000	300	30	5	20	50	230	300	10	30	5
,,	BD	215,000	300	30	4	230	50	20	300	10	30	4
,,	BC	130,000	190	19	3	130	40	20	190	10	19	3
Riser 1	EF*fc*	30,000	50	12	2	30	20	—	50	4	12·5	2
	FG*gf*	10,000	50	3	1	24	18	1	43	16	2·7	1
Branch	F*f*	20,000	50	12	$1\frac{1}{2}$	6	15	11	32	4	8	$1\frac{1}{2}$
Riser 2	—	28,000	50	5	$1\frac{1}{2}$	24	16	—	40	10	4	$1\frac{1}{2}$
,, 3	—	20,000	50	10	$1\frac{1}{2}$	16	20	—	36	5	7·2	$1\frac{1}{2}$
1	2	3	4	5	6	7	8	9	10	11	12	13

$\frac{50}{4}$ giving a ratio of 12 approximately. For the latter value, Chart 15 gives a 2-inch diameter pipe to supply the 30,000 B.Th.U., which are demanded by the first portion of the riser.

Use the same circuit length for FG*gf*, but the height given is 16 feet. Roughly, $\frac{50}{16} = 3$, and as this section must carry 10,000 B.Th.U., Chart 15 shows that a 1-inch diameter pipe will be necessary.

For the branch F*f*, the length may also be taken as 50 feet, whilst the circuit height is 4 feet. The ratio roughly is 12, and from the same chart a $1\frac{1}{2}$-inch diameter pipe is shown to be suitable for supplying 20,000 B.Th.U.

The remaining risers are treated in the same way. From

a superficial observation, it will be clear that the foregoing method will not equalize the resistance of the different branches, so the more refined method is continued to attain this end.

Beginning again with Riser No. 1 (Fig. 144) and observing the tabulated particulars, the total length for EF*fe* comes out at 50 feet, and although the correct ratio is 12·5 this does not alter the size of the pipe. In fact, corrections which only diminish or increase the length by a few feet, do not as a rule alter the size of the piping. They, however, show the actual state of affairs, and indicate where a certain amount of throttling is desirable, which process may be effected by simple forms of tee fittings with obstruction plugs, or by reducing the sizes of the radiator valves.

Dealing now with the section FG*gf*, this will be found to have a net length of 24 feet, to which 18 feet are added for the resistance of radiator and fittings, and an additional foot for the proportional resistance of the lower section. The latter is added in virtue of the 10,000 B.Th.U. having first to pass through EF, and it is estimated by the aid of Table XXIV. in the appendix. On reference to the table, the proportional resistance of a 2-inch diameter pipe in terms of 1-inch diameter piping is 0·024. This is multiplied by the total length of 50 feet which is given in column 10 opposite EF*fe*. Fractional quantities are omitted from the totals.

The branch F*f* has a net length of 6 feet, and 15 feet are added for the equivalent of fittings. Referring to Table XXIV., the proportional resistance per foot of the lower part of riser when expressed in 1½-inch diameter pipe is 0·21 foot, and 50 × 0·21 = 10·5, or say 11 feet. The latter value is noted in column 9. For the branch, a total length of 32 feet is obtained, and this when divided by the circuit height gives 8. Chart 15 will show that for a ratio of 8 the diameter must remain unchanged. All the sections of Fig. 144 excepting BC have a capacity in excess of their requirements.

Method of Sizing "Two-Pipe Up-Feed" Systems.—For this purpose Fig. 145 is used, and the particulars are tabulated as before. Assume a temperature drop in the mains of 30° F. and an average for the risers of 25° F.

To obtain the approximate sizes of mains, take the net length of the longest circuit, which for the case in hand, is say, 300 feet. The effective circuit height for the mains is the vertical distance from the centre of the average lowest radiator to the boiler fire grate, and is indicated by h_1 in Fig. 145. For the risers, their effective heights are represented by $h_1 h_2 h_3$ and h_4, these being the vertical distances from centre of heating surfaces to fire grate of the boiler.

The section ABGH is required to supply 124,000 B.Th.U.

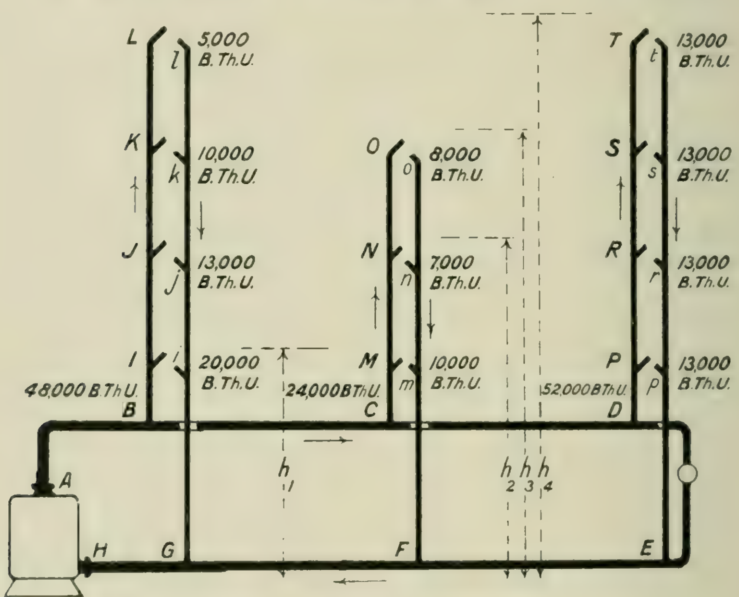


FIG. 145.—"Two-pipe up-feed" system.

for a circuit height of 12 feet. As the length of the longest circuit is 300 feet, then $\frac{300}{12} = 25$, and, according to Chart 11, a 3-inch diameter pipe will just suffice. Section BCFG must carry 76,000 B.Th.U., and for a ratio of 25, a $2\frac{1}{2}$-inch diameter pipe is found to have about the right capacity.

Section CDPpE supplies 52,000 B.Th.U., and for a ratio of 25, the $2\frac{1}{2}$-inch diameter pipe is carried forward.

In checking the sizes of the mains of Fig. 145, the full length of 300 feet is taken for the section ABGH, and to this

are added 36 feet for the resistance of fittings and boiler connections, these making a total of 336 feet. For a ratio of 28, a 3-inch diameter pipe is found to be a little too small, but it may be adopted.

The section BCFG is measured from B and G to the far end

PARTICULARS OF FIG. 145.

		Rough method.				More refined method						
Part of system.	Section.	Capacity in British Thermal Units.	Trial length of circuit l.	Approx. value of $\frac{l}{h}$	Trial dia. in ins d	Net length of section.	+ Equivalent length for fittings, etc.	+ Proportional length of circuit in terms of section required	= Total length of circuit l.	Effective circuit height h.	Length of circuit divided by height $\frac{l}{h}$.	Corrected diameter in inches d.
Main	ABGH	124,000	300	25	3	60	36	240	336	12	28	3
,,	BCFG	76,000	300	25	2½	100	15	176	291	12	24	2½
,,	CDP*p*E	52,000	300	25	2½	170	25	106	301	12	25	2½
Riser 1	PR*rp*	39,000	55	2·3	2	24	12	20	56	24	2·4	1½
,,	RS*sr*	26,000	55	1·5	1¼	24	4	22	50	36	1·4	1¼
,,	ST*ts*	13,000	55	1·0	1	24	10	4	38	48	0·79	¾
Branch	P*p*	13,000	55	4·6	1¼	8	18	3	29	12	2·4	1*
,,	R*r*	13,000	55	2·3	1	8	14	2	24	24	1·0	¾
,,	S*s*	13,000	55	1·5	1	8	10	4	22	36	0·6	¾*
Riser 2	CM*m*F	24,000	60	5·0	1½	28	12	6	46	12	3·8	1¼
,,	MN*nm*	15,000	60	2·5	1	24	6	14	44	24	1·8	1
,,	NO*on*	8,000	60	1·6	¾	24	8	9	41	36	1·1	¾*
Branch	M*m*	10,000	60	5·0	1	8	14	13	35	12	2·9	1*
,,	N*n*	7,000	60	2·5	¾	8	10	9	27	24	1·1	¾*
Riser 3	BI*i*G	48,000	55	4·6	2	25	16	11	52	12	4·3	2
,,	IJ*ji*	28,000	55	2·3	1¼	24	4	4	32	24	1·3	1¼
,,	JK*kj*	15,000	55	1·5	1	24	2	10	34	36	0·94	1
,,	KL*lk*	5,000	55	1·0	¾	24	8	1	33	48	0·69	½
Branch	I*i*	20,000	55	4·6	1¼	8	18	4	30	12	2·5	1¼
,,	J*j*	13,000	55	2·3	1	8	14	10	32	24	1·3	1*
,,	K*k*	10,000	55	1·5	¾	8	10	8	26	36	0·72	¾*
1	2	3	4	5	6	7	8	9	10	11	12	13

of the circuit, and it is 240 feet in length. To this is added, say 15 feet for fittings and the proportional resistance of the section ABGH. Expressing the resistance of the (60 + 36) = 96 feet of 3-inch diameter pipe in terms of the 2½-inch size, Table XXIV.

* Outlet connections should be throttled to reduce capacity.

in appendix gives 0·375 as the proportional resistance of 1 foot of 3-inch diameter pipe. Then $0·375 \times 96 = 36$ feet, giving a total length of $240 + 15 + 36 = 291$ feet, which divided by $12 = 24$. Chart 11 shows that for this ratio, a $2\frac{1}{2}$-inch diameter pipe is required.

Section CDP*p*E takes in the first portion of the end riser, and is measured from C and F. To this are added 25 feet for fittings, etc., and the proportional resistance of the previous sections. As the second section BCFG is of the same bore, seven-tenths of this may be added, whilst the 96 feet of 3-inch diameter piping are equal to 36 feet of $2\frac{1}{2}$-inch bore, so far as frictional resistance is concerned. Here the total length is $170 + 25 + 70 + 36 = 301$ feet, which gives a ratio of 25. The $2\frac{1}{2}$-inch diameter pipes as provisionally fixed upon would be retained, but a portion of its length may be diminished to the next smaller size. If the latter is adopted, it will be essential to calculate the proportional resistance in terms of the smaller pipe, and to see that the necessary capacity is maintained.

The extent to which refinements in calculations may be carried in practice is regulated by the economic side of the question, and few would dream of carrying them beyond the point where the financial advantage begins to disappear.

Risers of "Two Pipe" Systems.—To obtain the approximate length for sizing the risers, measure the length of the flow and return to the largest radiator, and add 30 feet.

Assume Riser No. 3 is being sized. Section BI*i*G must carry 48,000 B.Th.U., and its net length is 25 feet. With 30 feet added, and with a circuit height of 12 feet, the ratio is 4·6, and using Chart 15, a 2-inch diameter pipe is selected.

Section IJ*ji* carries 28,000 B.Th.U., and for a height of 24 feet, gives a ratio 2·3. From the same chart, a $1\frac{1}{4}$-inch diameter pipe is chosen.

Section JK*kj* is 36 feet high, and gives a ratio of 1·5. To supply the 15,000 B.Th.U. a 1-inch diameter pipe is found to be of a liberal size.

Section KL*lk* must carry 5000 B.Th.U., and for a circuit height of 48 feet this gives a ratio of over 1. For this a $\frac{3}{4}$-inch diameter pipe is selected from Chart 15.

The branches I*i*, J*j*, and K*k*, supply 20,000, 13,000, and

10,000 B.Th.U. respectively, and for the heights given, require $1\frac{1}{4}$-inch, 1-inch, and $\frac{3}{4}$-inch diameter connections.

Instead of adopting the above method for ascertaining the approximate sizes of the risers, the values could have been taken from Table XVI.

The sizes for this riser may now be checked by the more refined method.

Section BI*i*G has a net length of 25 feet, to which 16 feet are added for fittings, and 11 feet for the proportional resistance of that part of the main circuit indicated by ABGH. Table XXIV. in appendix gives the proportional resistance of 3-inch diameter pipe in terms of the 2-inch diameter branch as 0·115, and $(60 + 36) \times 0{\cdot}115 = 11$ feet. The total length, therefore, to be credited to BI*i*G $= 25 + 16 + 11 = 52$ feet, which divided by 12 gives 4·3. For this ratio, no change of size is made.

Section IJ*ji* is 24 feet in length, to which are added the equivalent of fittings and the proportional resistance of the circuit from the boiler to point J. The resistance of the 3-inch diameter main has already been found to be equal to that of 11 feet of 2-inch diameter piping. This gives a length of 2-inch diameter piping to points I*i* of 52 feet. The proportional resistance of 2-inch piping in terms of $1\frac{1}{4}$-inch tube is 0·081, and $52 \times 0{\cdot}081 = 4{\cdot}2$ feet. The total length of IJ*ji* now becomes $24 + 4 + 4 = 32$ feet, and as this gives a ratio of 1·3, a $1\frac{1}{4}$-inch diameter pipe is still necessary to supply the 28,000 B.Th.U. demanded.

The total resistance of section IJ*ji*, when estimated for $1\frac{1}{4}$-inch diameter piping, has been found as equal to 32 feet, and this, when estimated as the equivalent of 1-inch diameter piping, will be $32 \times 0{\cdot}3 = 9{\cdot}6$, or say 10 feet. The total length of section JK*kj* is therefore $24 + 2 + 10 = 34$ feet, and $\frac{34}{36} = 0{\cdot}94$. The latter ratio shows that a 1-inch diameter pipe is rather large, but this size would probably be adopted.

On account of the section KL*lk* having a capacity greatly in excess of the requirements, a smaller size will be tried. As the proportional resistance to K*k* was equal to 34 feet of 1-inch piping, this in turn is equivalent to $34 \times 0{\cdot}027 = 0{\cdot}92$, or say 1 foot of $\frac{1}{2}$-inch diameter pipe. The length of the top section of riser No. 3 is therefore $24 + 8 + 1 = 33$ feet, and $\frac{33}{48} = 0{\cdot}69$

For this ratio, Chart 15 shows that a $\frac{1}{2}$-inch diameter pipe is about the right size to supply 5000 B.Th.U.

The connections may now be dealt with in the same manner. To Ii, the proportional resistance is $52 \times 0{\cdot}081 = 4{\cdot}2$ feet of $1\frac{1}{4}$-inch diameter pipe, and $8 + 18 + 4 = 30$ as the total length. For a height of 12 feet, the ratio is 2·5, and for supplying 20,000 B.Th.U. the first diameter would be used.

For connection Jj, the proportional resistance is equal to 32 feet of $1\frac{1}{4}$-inch diameter pipe, and expressing this in terms of 1-inch diameter piping, the length is equal to $32 \times 0{\cdot}3 = 9{\cdot}6$, or say 10 feet. Thus the total length of Jj is $8 + 14 + 10 = 32$ feet, and $\frac{32}{24} = 1{\cdot}3$. For this ratio, a 1-inch diameter pipe is rather large, but further retardation may be introduced by using a $\frac{3}{4}$-inch radiator valve, whilst finer adjustment can be effected by throttling the outlet connection.

The proportional resistance for Kk in terms of $\frac{3}{4}$-inch diameter pipe is equal to 7·5 feet, giving a total length of $8 + 10 + 8 = 26$ feet. This gives a ratio of 0·72, but a $\frac{3}{4}$-inch diameter pipe would require to be restricted. For the remaining risers, the same method is followed.

Method of Sizing Pipes of "Down-feed" Systems.—Let Fig. 146 be the system under review, the sizing of which is for a temperature drop of 40° F.

After the general design has been prepared and the lengths of the pipes obtained, it is necessary to ascertain the effective circuit height. This can only be approximated in "down-feed" systems, as the conditions differ when compared with those of up-feed installations. If, however, the correct circuit height is not adopted, it will be discovered when making the refined calculations. It will be observed, for example, that the highest radiators of Fig. 146 are subject to a height of h_4, and the lowest to a height of h_1. Now, it is clear, that if the upper portion of a riser, say CN, is sized for a circulating height of h_4, too small a pipe will be given to serve effectively the lower section PQ. In like manner, if the whole of the "drop" pipe is figured for a circuit height of h_1 the upper part will be unnecessarily large. The mean height will probably be the best to take, although this may be adjusted according to the height where the maximum heat is emitted. In Fig. 146, all the

pipes at the outset are sized for the mean circuit height h_m, whilst the branches and returns are corrected in accordance with the height to which they are subjected.

For the first approximation in sizing the principal mains, take the longest circuit, which in Fig. 146, is ABEIT, and measures 402, or say 400 feet. The latter value and a mean circuit height of 25 feet give a ratio of 16, and according to

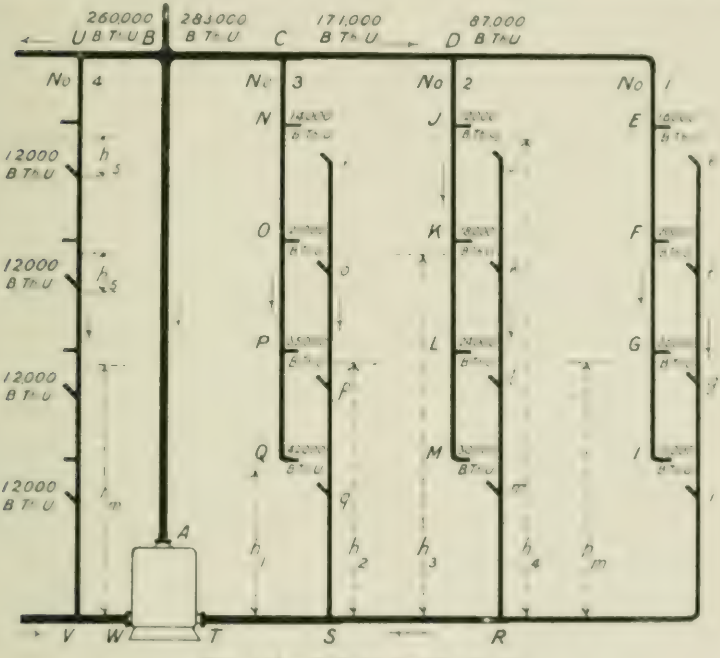


FIG. 146.—"Down-feed" system.

Chart 14, AB should be of $4\frac{1}{2}$ inches diameter, BC and TS $3\frac{1}{2}$ inches diameter, CD and SR 3 inches diameter, and DE along with iR 2 inches diameter.

These sizes may now be checked by the more exact method. The full length of circuit is taken for AB, so that no change is effected here.

Sections BC and TS have a length of 60 feet, to which are added the equivalent for fittings, the remainder of the circuit, and the proportional resistance of AB. The latter value is

PARTICULARS OF FIG. 146.

Part of system.	Section.	Capacity in British Thermal Units.	Trial length of circuit l.	Approx. value of $\frac{l}{h}$.	Trial dia. in ins. d.	Net length of section +	Equivalent length for fittings, etc. +	Proportional length of circuit in terms of section required =	Total length of circuit l.	Effective circuit height h.	Length of circuit divided by height $\frac{l}{h}$.	Corrected diameter in inches d.
Main . .	AB	543,000	400	16	4½	70	54	332	456	25	18	4½
,, . .	BC, TS	283,000	400	16	3½	60	40	301	401	25	16	3½
,, . .	CD, SR	171,000	400	16	3	80	20	250	350	25	14	3
,, . .	DE, iR	87,000	400	16	2½	150	20	54	224	25	9	2
,, . .	BU, VW	260,000	400	16	3½	40	60	340	440	25	17·5	3½
Drop . .	EF	71,000	150	6	1½	12	10	58	80	25	3·2	1½
Pipe 1 . .	FG	51,000	150	6	1½	12	10	42	64	25	2·5	1¼
	GI	15,000	150	6	1	12	10	10	32	25	1·3	¾
	E*ef*	16,000	150	3	¾	20	10	14	44	50	0·88	¾*
	fg	36,000	150	4	1¼	12	10	16	38	38	1	1
	gi	72,000	150	6	1½	12	10	64	86	26	3·3	1½
Connection	F*f*	20,000	150	4	1	8	10	8	26	38	0·68	¾*
,,	G*g*	36,000	150	6	1¼	8	10	20	38	26	1·4	1*
,,	I*i*	15,000	150	11	1	8	10	32	50	14	3·5	¾
Drop . . Pipe 2 . .	DJ, *m*R	84,000	140	5·6	2	35	20	22	77	25	3	1½
	JK	72,000	140	5·6	1½	12	20	66	98	25	3·9	1½
	KL	54,000	140	5·6	1½	12	20	43	75	25	3	1¼
	LM	30,000	140	5·6	1¼	12	20	22	54	25	2·1	1
	J*jk*	12,000	140	2·8	¾	20	10	12	42	50	0·8	½
	kl	30,000	140	3·7	1¼	12	4	8	24	38	0·63	¾
Connection	*lm*	54,000	140	5·4	1½	12	4	22	38	26	1·4	1¼
,,	K*k*	18,000	140	3·7	1	8	10	7	25	38	0·65	¾*
,,	L*l*	24,000	140	5·4	1	8	10	6	24	26	0·92	¾
,,	M*m*	30,000	140	10	1¼	8	15	54	77	14	5·5	1¼
Drop . . Pipe 3 . .	CN, *q*S	112,000	114	4·5	2	35	20	25	80	25	3·2	2
	NO	98,000	114	4·5	2	12	20	29	61	25	2·4	1½
	OP	77,000	114	4·5	1½	12	20	50	82	25	3·2	1½
	PQ	42,000	114	4·5	1¼	12	20	31	63	25	2·5	1¼
	N*no*	14,000	114	2·3	¾	18	10	12	40	50	0·8	¾*
	op	35,000	114	3	1¼	12	4	13	29	38	0·76	1
	pq	70,000	114	4·4	1½	12	4	31	47	26	1·8	1¼
	O*o*	21,000	114	3	1	8	10	6	24	38	0·63	¾*
	P*p*	35,000	114	4·4	1¼	8	10	10	28	26	1·0	1*
	Q*q*	42,000	114	8	1½	8	15	63	86	14	6	1¼
Drop 4 . .	UV	48,000	108	4·3	1¼	80	14	1	95	25	3·8	1¼
Radiator connection		12,000	20	10	1	8	10	—	18	2	9	1*
1	2	3	4	5	6	7	8	9	10	11	12	13

* Outlet connections should be throttled to reduce capacity.

(70 + 54) × 0·255 = 31 feet, and the total length for the sections will be 60 + 40 + 31 + 270 = 401 feet. The fraction 0·255 is the proportional value from Table XXIV. in appendix. In this case, a ratio of 16 is obtained, so a $3\frac{1}{2}$-inch diameter pipe is still required.

Sections CD and SR have a net length of 80 feet, to which are added the equivalent for fittings, the proportional resistance to C, and the length DEI*i*R. As shown above, the resistance of the $4\frac{1}{2}$-inch diameter pipe is equal to 31 feet of $3\frac{1}{2}$-inch pipe, and when BC and TS are added, amounts to (31 + 60 + 40) = 131 feet. Expressing this in terms of 3-inch diameter pipe, 131 × 0·44 = 57·6, say 58 feet. The total length for these sections becomes 80 + 20 + 58 + 192 = 350 feet, and when divided by 25 gives 14. For this ratio a 3-inch diameter pipe is used to supply the 171,000 B.Th.U. demanded.

As the main pipes already checked have very liberal capacities, a 2-inch diameter pipe may be tried for the sections DE and *i*R. The total length of 3-inch pipe as estimated above amounts to (80 + 20 + 58) = 158 feet, and 158 × 0·115 = 18 feet of 2-inch diameter pipe, which offers the same resistance. For these sections the total length is 150 + 20 + 18 + 36 = 224 feet, and divided by 25 = 9. The latter ratio, according to Chart 13, shows the smaller pipe to be of ample size.

"Drop" Pipes.—For approximating the size of the "drop" pipes use the length of the section under consideration, and add to it one-third of the length of the main circuit up to that section. Thus the trial length for E to I*k* will be $36 + \frac{354}{3}$ = 154, or say 150 feet. Drop pipe No. 2 will have a trial length of DM*m*R plus one-third of ABDRT, and gives $71 + \frac{210}{3}$ = 141, or say 140 feet. For "drop" No. 3 the trial length will be $71 + \frac{129}{3}$ = 114 feet. Use the mean circuit height h_m for sizing the supply pipes, whilst for the radiator connections and returns use the heights by which these are influenced directly.

When radiators are joined with a single pipe as "drop," No. 4, the average circuit height, may be used, but a higher or lower value may sometimes be necessary, according to whether the greater portion of the heating surface is concentrated near to the top or to the bottom of the "drop" pipe.

For "drop" pipes which resemble No. 4, the radiator connections have an effective circuit height of h_5, which is, roughly speaking, the vertical distance from the centre of the radiator to the lowest part of the return connection.

In checking the trial values when the "drop" pipes are on the "two-pipe" principle, as No. 1, estimate the proportional resistance of the circuit to which they are joined, and add half the length of the return "drop" pipe. EF has a trial diameter of $1\frac{1}{2}$ inches. Estimate the proportional resistance in terms of this size of the piping from A to E, and from i to T. This works out at 40 feet, to which is added half the length of *c to i*, or 18 feet, giving 58 feet as the proportional resistance. The total length for EF will be $12 + 10 + 58 = 80$ feet, and divided by $25 = 3{\cdot}2$. Chart No. 12 shows the $1\frac{1}{2}$-inch diameter pipe to be the right size.

The proportional resistance to FG $= 80 \times 0{\cdot}38 = 30{\cdot}4$ plus half *fi*, or 12 feet, giving 42 feet. The total length for the section is therefore $12 + 10 + 42 = 64$ feet, and divided by $25 = 2{\cdot}5$. This ratio gives a $1\frac{1}{4}$-inch diameter pipe.

Section GI has a proportional resistance of $64 \times 0{\cdot}067 = 4{\cdot}2$ feet, and with half *gi* added equals 10 feet. The total length here will be $12 + 10 + 10 = 32$ feet, and with a circuit height of 25 feet gives a ratio of $1{\cdot}3$.

The return "drops" are dealt with in the same manner, but the circuit heights are those which agree with the positions of the heating surfaces.

Finally, the radiator connections may be checked, the proportional resistance being estimated to each position.

The corrected diameter cannot always be obtained at one step, and a re-calculation is sometimes essential. The methods as given above, and also that used in conjunction with the previous figures, are continued with the remaining pipes and connections, whilst the tabulated values also indicate how the sizes are to be obtained.

CHAPTER XXI

SIZING PIPES FOR FORCED HOT-WATER CIRCULATING SYSTEMS

THE most economical circulating velocity depends largely upon the size of a plant and upon the efficiency of the circulating apparatus together with that of the prime mover. There is no convenient relationship between the cost of the various sizes of pipes, but the power to circulate water varies directly as the cube of the speed when the efficiency of the circulating apparatus remains unaltered. By estimating, however, the initial and operating costs of systems for high and low velocities the most suitable one can be determined.

For estimating the capacities of forced circulating systems the following formula may be used :—

$$W = c_1\sqrt{\frac{d^5 h}{l}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

where W = weight of water circulated per minute.

d = bore of pipe in inches.

h = circulating head in feet.

l = length of pipe in feet.

c_1 = variable coefficient as given below.

VALUES OF c_1.

Bore of pipe. Inches.	Value of c_1.	Bore of pipe. Inches.	Value of c_1.
[illegible]	175	$1\frac{1}{2}$	212
[illegible]	178	2	224
[illegible]	184	$2\frac{1}{2}$	[illegible]
[illegible]	192	3	240
1	198	$3\frac{1}{2}$	[illegible]
$1\frac{1}{4}$	206	4	255

The capacity of a circuit in terms of the velocity is given by Formula 50, or it may be taken from Chart 17.

$$W = 20d^2v \quad . \quad . \quad . \quad . \quad . \quad . \quad (50)$$

where v = velocity in feet per second.

W and d have the same meaning as in Formula 49.

The horse-power absorbed by pipe friction is estimated by the usual formula—

$$P_f = \frac{Wh}{33,000} \quad . \quad . \quad . \quad . \quad . \quad . \quad (51)$$

or, if the horse-power absorbed by the pump is required,

$$P_p = \frac{Wh}{33,000e} \quad . \quad . \quad . \quad . \quad . \quad . \quad (52)$$

or, if the power absorbed is required in terms of the size of a circuit and the weight of water circulated,

$$P_p = \frac{W^3l}{33,000c_1{}^2d^5e} \quad . \quad . \quad . \quad . \quad . \quad (53)$$

where P_f = horse-power absorbed by pipe friction.

P_p = " " by pump.

e = efficiency of pump.

The remaining symbols as in Formula 49.

Example 27.—Determine the thermal capacity of a simple loop circuit that has a diameter of 3 inches and a length of 500 feet, when the circulating speed is 6 feet per second, and the fall of temperature 20° F. What power would be absorbed by the pump in circulating the water if its efficiency is 40 per cent. ?

By Formula 50

$$W = 20d^2v$$

Substituting values, $W = 20 \times 3^2 \times 6$

when W = **1080 lbs. of water per minute.**

Chart 17 gives the same value.

Taking now the thermal capacity of the circuit per hour,

$$1080 \times 60 \times 20 = \mathbf{1,296,000\ B.Th.U.}$$

The horse-power absorbed by the pump is obtained by Formula 53, where

$$P_p = \frac{W^3 l}{33{,}000 c^2 d^5 e}$$

Substituting values, $P_p = \dfrac{1080^3 \times 500}{33{,}000 \times 240^2 \times 3^5 \times 0{\cdot}4}$

when P_p = **3·43 horse-power absorbed.**

Application of Chart to Loop Circuits.—To facilitate calculations in connection with forced circulation, the writer has prepared Charts 17 to 20, and it will be observed that in the three latter, the friction head and length of pipe are expressed as ratios. In order, therefore, to determine the head absorbed by a pipe in circulating any given weight of water, all that is necessary is to divide the length of piping by the ratio agreeing with the capacity under consideration. After having ascertained the head absorbed, the horse-power utilized may be determined by either Formula 52 or 53.

The thermal capacity of a circuit per hour is obtained by multiplying the weight of water circulated per minute by 60, and afterwards by the temperature through which the water falls.

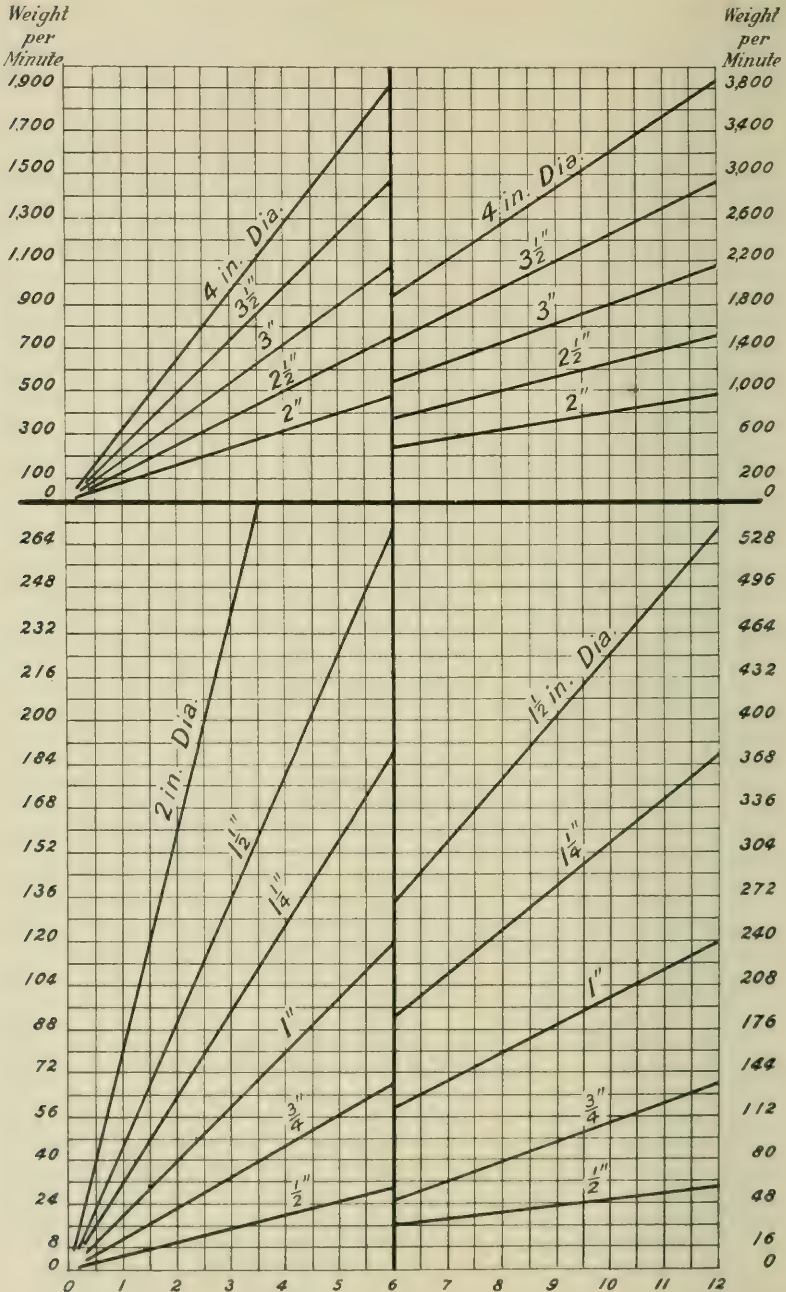
Application of Charts to "Two-Pipe" Systems.—At the outset the mains are sized in terms of the velocity by the aid of Chart 17, whilst to ascertain the head absorbed by friction either of the Charts 18 to 20 is used. The branches are sized directly from the charts according to the differential pressure that exists between the flow and the return pipes.

The method adopted by the writer is as follows:—

(1) Express the thermal capacity of all sections of a system in weight of water to be circulated per minute through them.

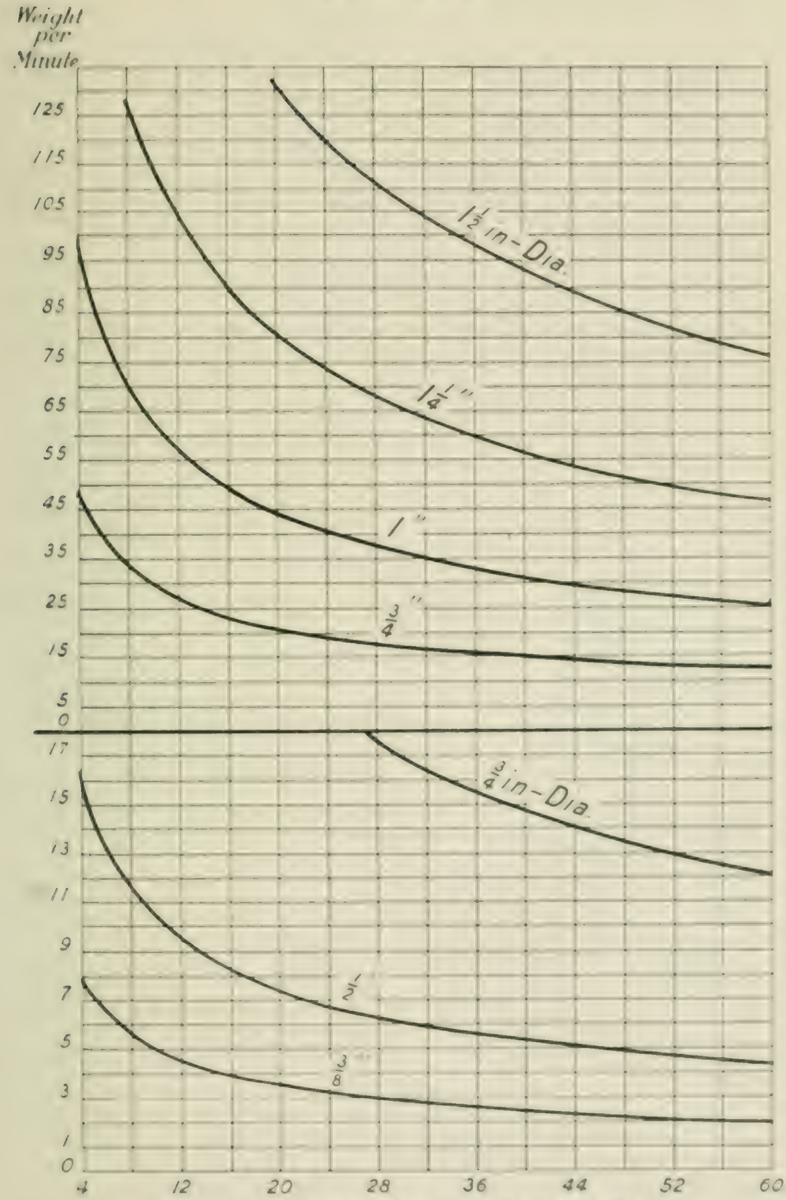
(2) Choose any arbitrary velocity for the mains and size them from Chart 17. Higher velocities should be chosen, as a rule, for the larger pipes than for those of a smaller bore, on account of their resistance being less for any unit rate of speed. If a velocity is chosen that is either too high or too low, it is revealed in calculating the loss of head, when corrections can be made accordingly.

CHART 17.—Forced hot-water circulation. Capacity of circuits in pounds of water per minute.



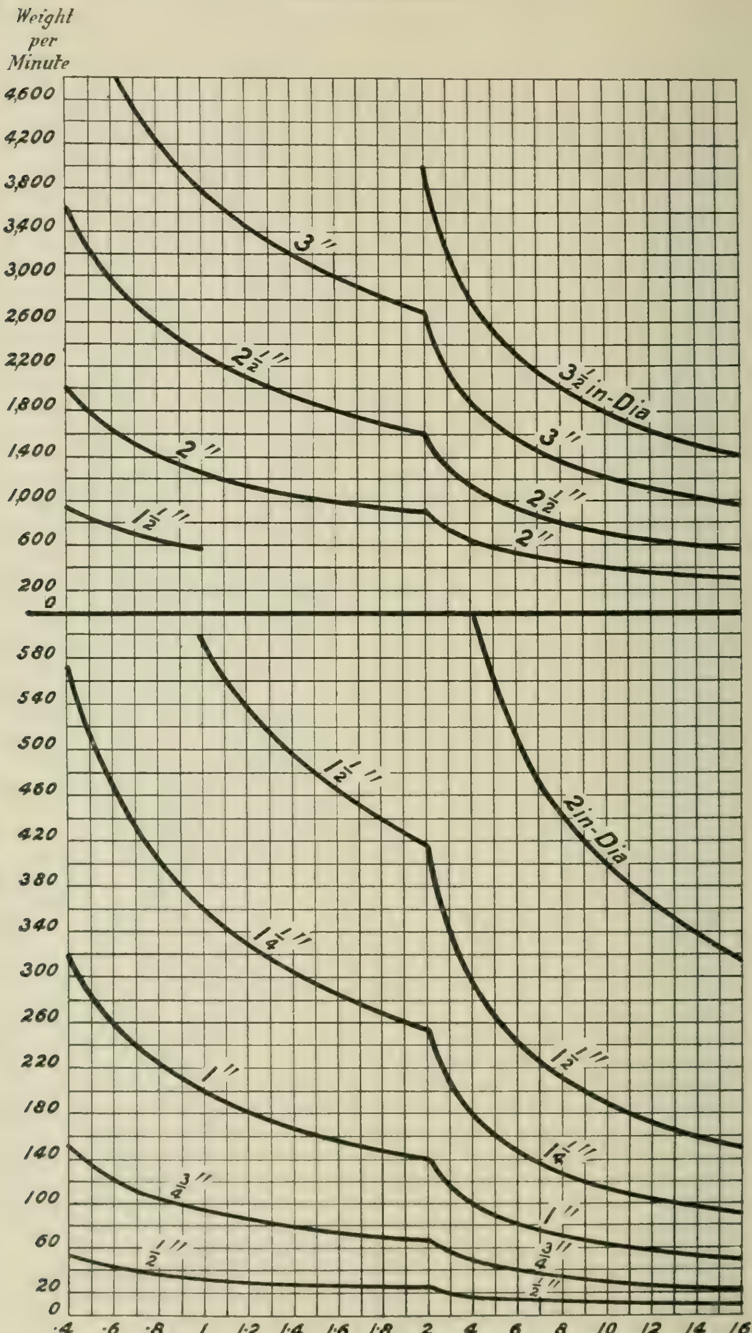
Velocity of flow in feet per second.

CHART 18.—Forced hot-water circulation. Capacity of circuits in pounds of water per minute.



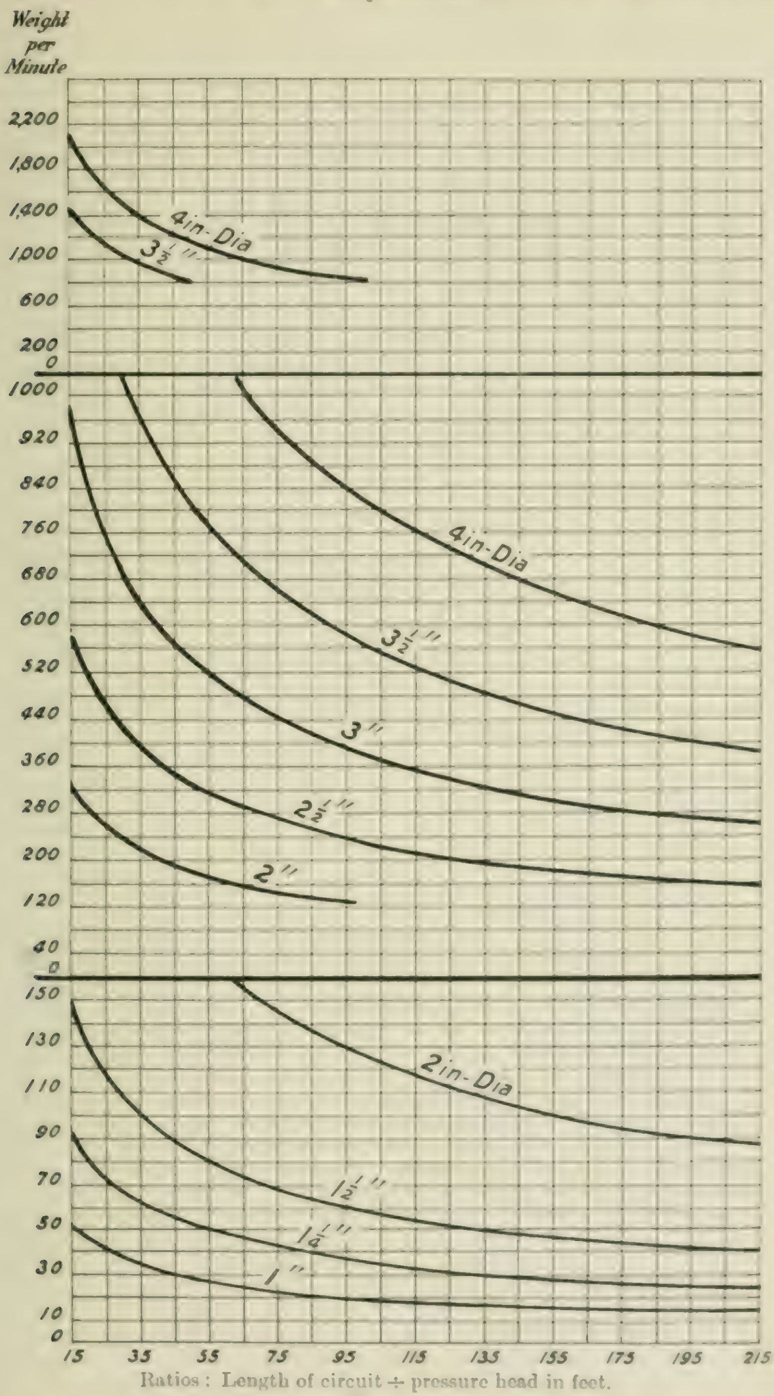
Ratios: Length of piping ÷ pressure head in feet.

CHART 19.—Forced hot-water circulation. Capacity of circuits in pounds of water per minute.



Ratios: Length of piping ÷ pressure head in feet.

CHART 20.—Forced hot-water circulation. Capacity of circuits in pounds of hot water per minute.



(3) Estimate the length of piping for each section, adding the equivalent length for the resistance of fittings.

(4) In dealing with the mains, read off from Charts 18 to 20, the values of the ratios $\frac{l}{h}$ that agree with the capacities and the sizes of pipes chosen.

(5) Estimate the frictional head for each section of the mains by dividing its total length by the ratios from the charts.

If now the heads absorbed for all the sections are added together, their sum will give the differential pressure head that should exist between the outlet and inlet of the pump. This

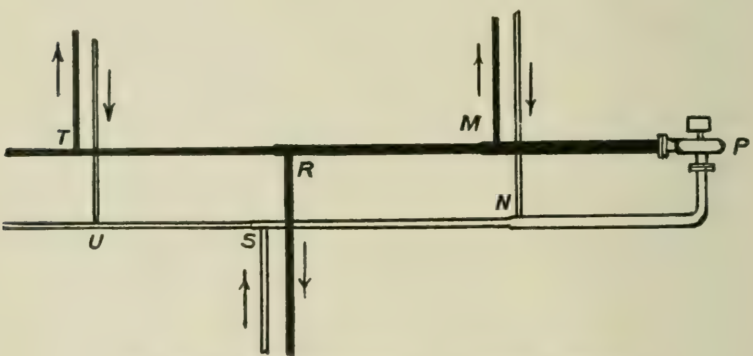


FIG. 147.—Forced circulation. "Two-pipe" system.

being done, the horse-power absorbed by the pump in circulating the water may be calculated by Formula 52.

After ascertaining the capacities and lengths of the branches, they are dealt with rather differently. In this case, the circulating head absorbed is computed to each of the branches on the main, when the differential head between the flow and return branches can be readily obtained.

The procedure then consists of dividing the length of a branch by the differential head available, and from the ratio obtained, the size is figured from the charts.

It will often be found that a standard size of pipe does not have the correct capacity, but in selecting a pipe that is rather large, the return piping should be throttled so as to pass just the weight of water required.

The following example will aid in making the explanation clear.

Example 28. Let the sizes of the piping be obtained for the arrangement as shown in Fig. 147, which represents a few separate buildings being supplied by one central plant. The length of the piping and capacities of the branches are given in the tabulated particulars below, whilst the temperature drop will be taken as 20° F. Let it be assumed that the loss of heat by the mains is included in the figures given. Estimate the horse-power absorbed by the pump if it has an efficiency of 55 per cent.

PARTICULARS OF FIG. 147.

Part of system.	Section.	Capacity in B.Th.U. per hour.	Temp. drop × 60	= Weight of water per minute.	Approx. velocity in feet per second.	Diameter of pipe in inches.	Length of section.	+ Equivalent length for fittings.	Total length of pipe.	Rate [illegible]	[illegible]
Main .	PM	1,500,000	1200	1250	7	3	60	20	80	9·2	8·7
,,	NP	,,	,,	,,	7	3	60	20	80	9·2	8·7
,,	MR	900,000	1200	750	6	2½	75	10	85	9·3	9·15
,,	SN	,,	,,	,,	6	2½	75	10	85	9·3	9·15
,,	RT	500,000	1200	417	5	2	100	10	110	9·1	12·1
,,	US	,,	,,	,,	5	2	100	10	110	9·1	12·1
,,	T to U	200,000	1200	167	3¾	1¼	260	20	280	12·4	22·6
Branches	MN	600,000	1200	500	—	1½*	40	15	55	0·85	65·1
	RS	400,000	1200	333	—	1½*	60	12	72	1·54	46·8
	TU	300,000	1200	250	—	1¼	40	10	50	2·2	22·6
1	2	3	4	5	6	7	8	9	10	11	12

Following along the lines as already explained, the pipes P to M and N to P will require to pass 1250 lb. of water per minute in order to yield 1,500,000 B.Th.U. per hour for the temperature drop given. For these sections, a velocity of 7 feet per second is chosen, which, with 3-inch diameter pipes, circulates the weight of water required. Each of the pipes PM and NP has a total length of 80 feet, and from Chart 19 it will be

* Denotes that return pipes should be throttled.

seen that a 3-inch diameter pipe when passing 1250 lb. per minute has a ratio of 9·2. Dividing 80 by 9·2, the head absorbed by each of these sections is found to be 8·7 feet.

The other sections for the mains are dealt with in the same manner as shown in the data above.

Before the branches can be sized, the circulating head at the different points is obtained, that at the inlet of the pump being taken as zero. It will be understood that the total pressure at the pump or at any other point is made up of the circulating plus the static head, the circulating head being that portion which is wholly absorbed in pipe friction.

On the above basis, and working backwards from the inlet of the pump, the circulating head at the different branches will be as follows :—

Part of system.	Circulating head in feet.	Part of system.	Circulating head in feet.
Pump inlet	0	Section T	54·55 includes 2 ft. for end of circuit.
Branch N	8·7	,, R	66·65
,, S	17·85	,, M	75·8
,, U	29·95	Pump outlet	84·5 total circulating head.

Let it be assumed that at the end of each branch in Fig. 147 a head of 2 feet is allowed for circulating the water through the pipes beyond. Upon this basis, the available head for sizing the branches will be as follows :—

Branches M and N, 75·8 − 8·7 − 2 = 65·1 feet
,, R and S, 66·65 − 17·85 − 2 = 46·8 ,,
,, T and U, 54·55 − 29·95 − 2 = 22·6 ,,

If now the lengths of the branches are divided by the heads at disposal, their sizes can be read off from Chart 19.

Considering the branches M and N, their combined lengths are given as 55 feet, and as the available head at this point is 65·1 feet, a ratio of 0·85 is obtained. Chart 19 will show that a $1\frac{1}{2}$-inch diameter pipe is the nearest size, but as its capacity would be too great, throttling will be essential to add the necessary resistance.

The power absorbed by the pump in 147 for the efficiency given is obtained by Formula 52, where

$$P_p = \frac{Wh}{33{,}000e}$$

Substituting values, $P_p = \frac{1250 \times 84 \cdot 5}{33{,}000 \times 0 \cdot 55}$

when $P_p = 5 \cdot 8$ horse-power.

Method of estimating Capacity of Branched Loops. For cases of this kind, the charts are not directly applicable, and

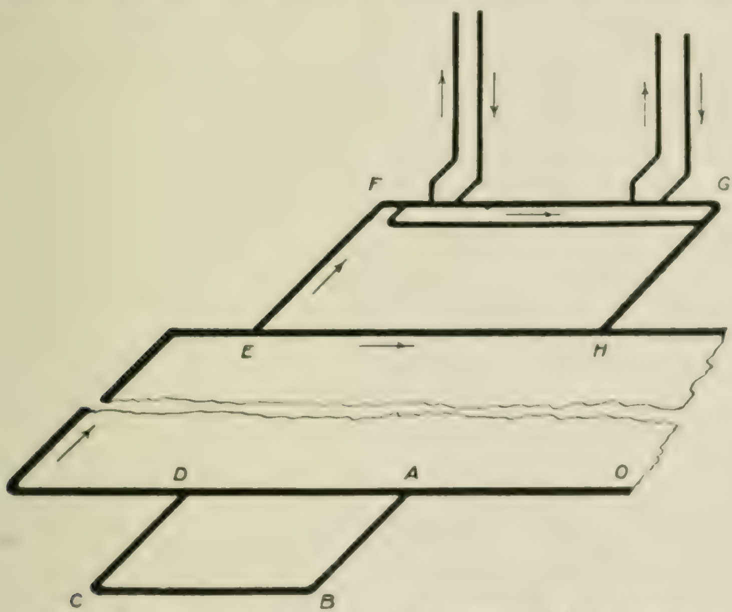


FIG. 148.—Forced circulated "one-pipe" system with loop circuits.

each example requires to be considered on its own bearings. Take, for example, Fig. 148, which has a main circuit OADEH, and to which are branched the loops ABCD and EFGH. When water is circulating, say, in the direction from O to H, two paths are provided at point A, and the relative weights that will flow through the main and the branched loops will be inversely proportional to their resistances.

Before endeavouring to estimate the capacity of a loop circuit, it is advantageous to express its proportional resistance in terms of the main circuit. This is readily done by the aid of Table XXIV. of appendix and the formula below.

$$l_p = \frac{lr_1}{n^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (54)$$

where l_p = proportional length of loop when its resistance is expressed in terms of the main circuit.

r_1 = proportional resistance of loop to main circuit as given in Table XXIV. of appendix.

l = length of loop in feet, including the equivalent resistance for fittings.

n = number of pipes forming loop.

If, however, the loop is formed of a single pipe, as ABCD, Fig. 148, then n is omitted, and

$$l_p = lr_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (54a)$$

On the other hand, if the loop is formed of single and double pipes, as in EFGH, each part is independently considered.

After having obtained the proportional length of a loop, its capacity may be calculated by the formula

$$W_1 = \frac{W\sqrt{L}}{\sqrt{l_p} + \sqrt{L}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

where W_1 = weight of water circulating per minute through branch loop.

W = total weight of water circulating through system per minute.

L = length in feet between junctions of loop and main as represented by AD and EH of Fig. 148.

l_p = as in Formula 54.

Example 29.—Let the main circuit of Fig. 148 be of 2-inch bore with a circulating speed at O of 5 feet per second; branch loop ABCD 30 feet long and of 1-inch bore, and the length between A and D 20 feet. The single piping of loop EFGH

has a length of 16 feet, and is of $1\frac{1}{4}$-inch bore, whilst the double pipe between FG is 50 feet in length and of $1\frac{1}{2}$-inch bore. The distance between EH is 50 feet, being the same as at FG. Assume the thermal capacity of these loops is desired in B.Th.U. per hour when the water falls through a temperature of 20° F.

Section ABCD.

(*a*) Add to the length of ABCD the equivalent for fittings, which is say, 20 feet, giving a total length of 30 + 20 = 50 feet.

(*b*) Obtain proportional resistance in terms of main circuit by the aid of Formula 54*a*, when

$$l_p = lr_1$$

It will be found that in Table XXIV. r_1 has a value of 41.

$$\text{when } l_p = 50 \times 41 = \mathbf{2050\ feet.}$$

(*c*) Obtain the capacity of the 2-inch diameter main circuit from Chart 17, which for 5 feet per second is 400 lb. per minute.

(*d*) The weight of water circulating through the loop is now determined by Formula 55, where

$$W_1 = \frac{W\sqrt{L}}{\sqrt{l_p} + \sqrt{L}}$$

Substituting values, $W_1 = \dfrac{400 \times \sqrt{20}}{\sqrt{2050} + \sqrt{20}}$

$$\text{when } W_1 = \mathbf{36\ lb.\ per\ minute.}$$

(*e*) Finally, the thermal capacity will be

$$36 \times 20 \times 60 = \mathbf{43{,}200\ B.Th.U.}\text{ per hour.}$$

Section EFGH.

(1) Add to the lengths of EF and GH the equivalent for fittings, which may be taken as 20 feet, giving a total length for these two pipes of 16 + 20 = **36 feet.**

The proportional resistance of $1\frac{1}{4}$-inch diameter pipe in terms of 2-inch diameter piping is given in Table XXIV. as 12·4, and 36 × 12·4 = **446 feet.**

Obtain proportional resistance of FG, which has a length of 50 feet, and with 10 feet added for fittings makes a total of **60 feet.**

The proportional resistance of the double pipe is found by Formula 54, where

$$l_p = \frac{lr_1}{n^2}$$

Expressing the $1\frac{1}{2}$-inch piping in terms of 2-inch diameter pipe, Table XXIV. shows $r_1 = 4{\cdot}7$.

Substituting values, $$l_p = \frac{60 \times 4{\cdot}7}{2^2}$$

when $l_p =$ **70 feet.**

The equivalent length of the whole of loop is therefore $446 + 70 =$ **516 feet.**

(2) Weight of water circulating through loop is now obtained by Formula 55.

$$W_1 = \frac{W\sqrt{L}}{\sqrt{l_p} + \sqrt{L}}$$

Substituting values, $$W_1 = \frac{400 \times \sqrt{50}}{\sqrt{516} + \sqrt{50}}$$

when $W_1 =$ **95 lb. per minute.**

(3) Thermal capacity for a temperature drop of 20° F.

$95 \times 20 \times 60 =$ **114,000 B.Th.U. per hour.**

The movement of water through the risers in Fig. 148 is governed largely by gravity circulation, and would be sized accordingly.

CHAPTER XXII

THE SIZING OF PIPES OF STEAM-HEATING SYSTEMS

General Formulæ.—The permissible velocity of steam in the pipes of a heating apparatus depends upon the following considerations:—

(1) The form of the apparatus.

(2) The flow of the condensation and the steam whether in the same or in opposite directions.

(3) The vertical distance between the boiler and the lowest heating surfaces that are joined with the gravity returns.

In low-pressure steam heating formulæ, it is not usual to include any function for the variable velocity that occurs through differences of pressure, and this is scarcely necessary owing to the comparatively small pressure drop that is allowed.

As a general formula for determining the capacity of steam pipes, the writer gives the following:—

$$U = c_2 L \sqrt{\frac{d^5 p D}{l}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

When transposed,

$$d = \sqrt[5]{\left(\frac{U}{c_2 L}\right)^2 \times \frac{l}{pD}} \quad . \quad . \quad . \quad . \quad . \quad (57)$$

and

$$p = \left(\frac{U}{c_2 L}\right)^2 \times \frac{l}{d^5 D} \quad . \quad . \quad . \quad . \quad . \quad . \quad (58)$$

where U = capacity in British Thermal Units per hour.

L = latent heat of steam.

D = density of steam in lb. per cubic foot.

d = diameter of pipe in inches.

where p = drop of pressure in ounces per square inch.
l = length of pipe in feet.
c_2 = coefficient from Table XVII.

For values of L and D see Steam Table in appendix.

Should the pressure drop be required in inches of water,

then $$U = 0{\cdot}76c_2L\sqrt{\frac{d^5h_1D}{l}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (59)$$

and $$d = 1{\cdot}12\sqrt[5]{\left(\frac{U}{c_2L}\right)^2 \times \frac{l}{h_1D}} \quad . \quad . \quad . \quad (60)$$

and $$h_1 = 1{\cdot}73\left(\frac{U}{c_2L}\right)^2 \times \frac{l}{d^5D} \quad . \quad . \quad . \quad . \quad . \quad . \quad (61)$$

where h_1 = pressure drop in inches of water-gauge.

The remaining symbols as before.

TABLE XVII.

VALUES OF c_2.

Diameter in inches.	Value of c_2.	Diameter in inches.	Value of c_2.
$\frac{1}{4}$	530	$2\frac{1}{2}$	700
$\frac{3}{8}$	540	3	720
$\frac{1}{2}$	550	$3\frac{1}{2}$	750
$\frac{3}{4}$	580	4	770
1	610	$4\frac{1}{2}$	780
$1\frac{1}{4}$	620	5	790
$1\frac{1}{2}$	640	6	820
2	670	8	850

Conditions where only Limited Steam Velocities are Applicable. —Formulæ 56 to 61 are only for cases where the water of condensation is conveyed by a separate pipe, or when it flows in the same direction as the steam. Where, however, the condensation and steam flow through the same pipe but in opposite directions, the velocity of the steam will require to be limited in order to avoid an excessive wave motion which would interfere with the steam supply.

Fig. 149 will illustrate what is meant. Let it be assumed that the pipe has a pitch just sufficient to cause the condensation to gravitate in the direction shown, the only reaction being

that of pipe friction. It will be clear that a certain amount of wave motion will be caused by the opposing stream of steam as indicated in the figure, the height of the wave varying with the velocity of the steam. In some cases, where the velocity is excessive, the condensation is held back until a sufficient head is created to overcome the resistance, or in others, it may be dislodged to some other part of the system where relief is effected by means of a "drip." Under such conditions, more or less hammering is caused.

In low-pressure systems, the height of the wave motion that

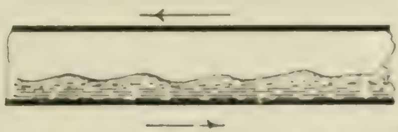


FIG. 149.—Steam and condensation flowing in opposite directions.

is caused by the opposing currents of condensation and steam may be obtained approximately by the following formula :—

$$h_w = \frac{v^2}{7740} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (62)$$

$$\text{from which } v = 88\sqrt{h_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad (63)$$

where h_w = height of wave motion in inches.

v = velocity of steam in feet per second.

Example 30.—Assume the maximum velocity is required where it is desired that the wave motion shall not exceed half an inch.

By Formula 63—

$$v = 88\sqrt{h_w}.$$

Substituting values, $v = 88\sqrt{0{\cdot}5}$

when v = 62 feet per second.

It is only, of course, in large pipes where half an inch of wave action could be allowed, but in order to have uniformity it may be expressed as a fraction of the pipe size. Thus, if the wave action is limited in height to one-sixteenth the bore of a pipe, then for diameters ranging from 1 to 8 inches, the

maximum velocities for the steam would vary from 22 to 62 feet per second.

Resistances of Bends and Fittings.—These are expressed in pipe equivalents as in Chapter XX., the various values being given in Table XXIII. of appendix.

In the two examples that follow, it is assumed that the resistance of pipe fittings is included in the length of pipe given.

Example 31.—Determine the capacity of a steam pipe 3 inches diameter, when its total length is 250 feet, the pressure at the boiler 2 lb. per square inch, and when a fall of pressure of 6 oz. per square inch can be allowed between the ends of the pipe.

By Formula 56—

$$U = c_2 L \sqrt{\frac{d^5 p D}{l}}.$$

For a 3-inch diameter pipe $c_2 = 720$, whilst for a gauge pressure of 2 lb. per square inch L and D may be considered as equal to 966 and 0·0427 respectively.

Substituting values, $$U = 720 \times 966 \sqrt{\frac{3^5 \times 6 \times 0{\cdot}0427}{250}}$$

when $U =$ **347,000 B.Th.U.** per hour.

Example 32.—A 2-inch diameter steam pipe 100 feet long supplies 80,000 B.Th.U. per hour. Determine the drop of pressure in inches of water for an initial steam pressure of 4 lb. per square inch.

For a 2-inch diameter pipe $c_2 = 670$, whilst the values of L and D for the pressure given may be taken as 962 and 0·0474.

By Formula 61—

$$h_1 = 1{\cdot}73 \left(\frac{U}{c^2 L}\right)^2 \times \frac{l}{d^5 D}.$$

Substituting values,

$$h_1 = 1{\cdot}73 \times \left(\frac{80{,}000}{670 \times 962}\right)^2 \times \frac{100}{2^5 \times 0{\cdot}0474}$$

when $h_1 =$ **1·75 inches of water pressure.**

Sizes of Return Pipes.—These are not derived as a rule by the aid of formulæ, but are governed by general considerations, such as the form taken by a system, length of piping, or according to the degree of freedom with which the air and condensation can be passed through the pipes.

The following table gives the sizes of returns suitable for general work :—

TABLE XVIII.

SIZES OF RETURN PIPES FOR STEAM HEATING SYSTEMS.

Diameter of steam pipes in inches.	Diameter of returns in inches.	
	Gravity systems (ordinary).	Atmospheric and Vacuum systems.
[illegible]	—	[illegible] *
$\frac{1}{2}$	$\frac{1}{2}$	[illegible] *
$\frac{3}{4}$	$\frac{1}{2}$	[illegible]
1	$\frac{3}{4}$	$\frac{1}{2}$
$1\frac{1}{4}$	1	$\frac{3}{4}$
$1\frac{1}{2}$	$1\frac{1}{4}$	1
2	$1\frac{1}{2}$	$1\frac{1}{4}$
$2\frac{1}{2}$	2	$1\frac{1}{4}$
3	$2\frac{1}{2}$	$1\frac{1}{2}$
$3\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{1}{2}$
4	3	2
$4\frac{1}{2}$	3	2
5	$3\frac{1}{2}$	2
6	$3\frac{1}{2}$	$2\frac{1}{2}$
8	4	3
9	$4\frac{1}{2}$	$3\frac{1}{2}$

Sizes of "Drip Pipes" or "Bleeders."—In iron pipe work, the sizes of the "drips" should not be less than [illegible] inch diameter on account of their liability to chokage through corrosion, whilst at the junctions of the pipes drained, the minimum diameter may, with advantage, be increased to half an inch. This size is large enough to drain a comparatively long pipe, but where large volumes of condensation occur, the outlet orifice may be calculated by Formula 64—

* Use next higher sizes for atmospheric systems.

$$d = \frac{Aw_3}{200} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

where d = bore of drip in inches at the junction of the pipe drained.

A = area of surface drained in square feet.

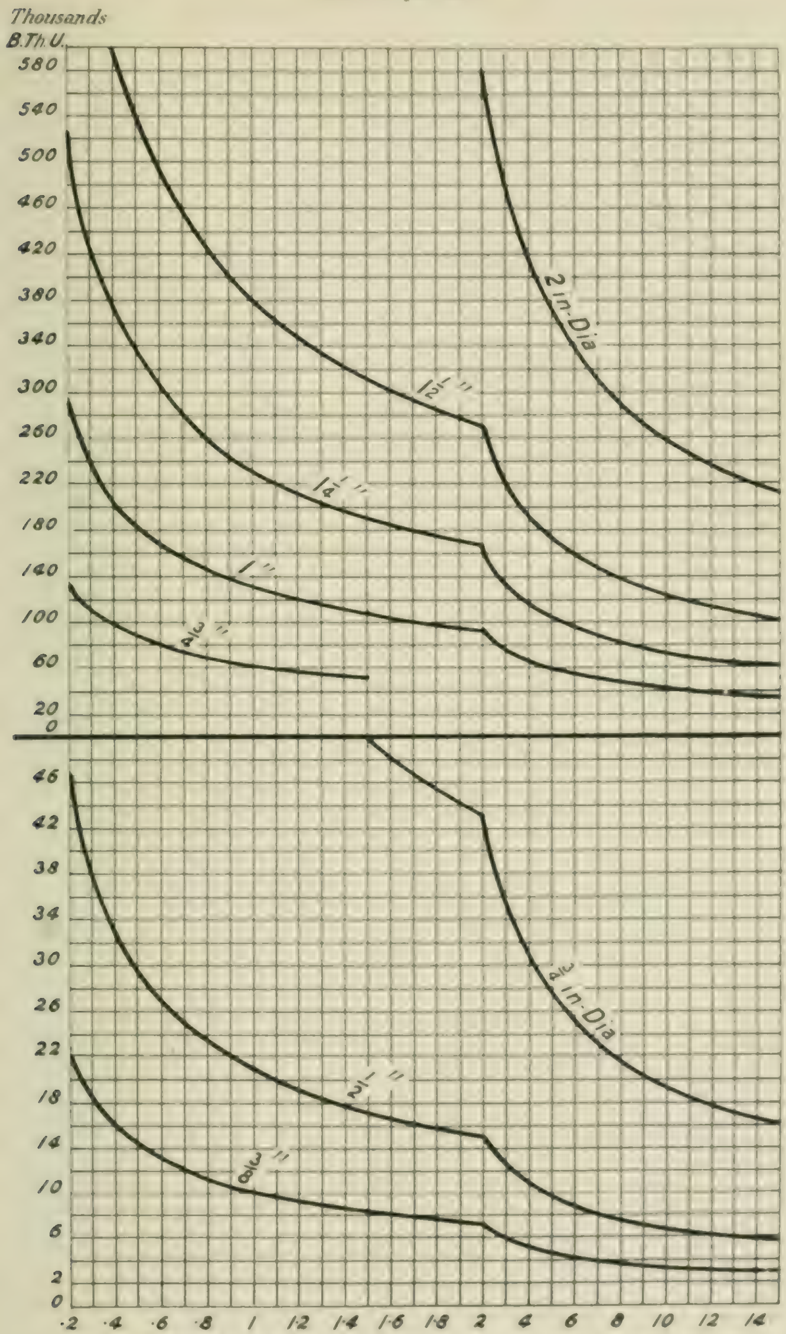
w_3 = weight of steam condensed in lbs. per square foot of surface per hour.

As the chief resistance in connection with drips occurs at their inlet orifices, the "drips" themselves may often be of a reduced bore.

Steam Charts.—For the expeditious sizing of steam pipes, the writer has prepared six charts, whilst a Table for ordinary Gravity "Two-Pipe" Systems is also given. Charts 21 to 23 are for an initial steam pressure of 5 lb. per square inch, where the drop of pressure is recorded in ounces per square inch. Charts 24 to 26 are for an initial steam pressure of 1 lb. per square inch, and wheie the pressure drop is given in inches of water. Along the bottom of the charts, ratios are given which are obtained by dividing the length of piping by the pressure drop at disposal.

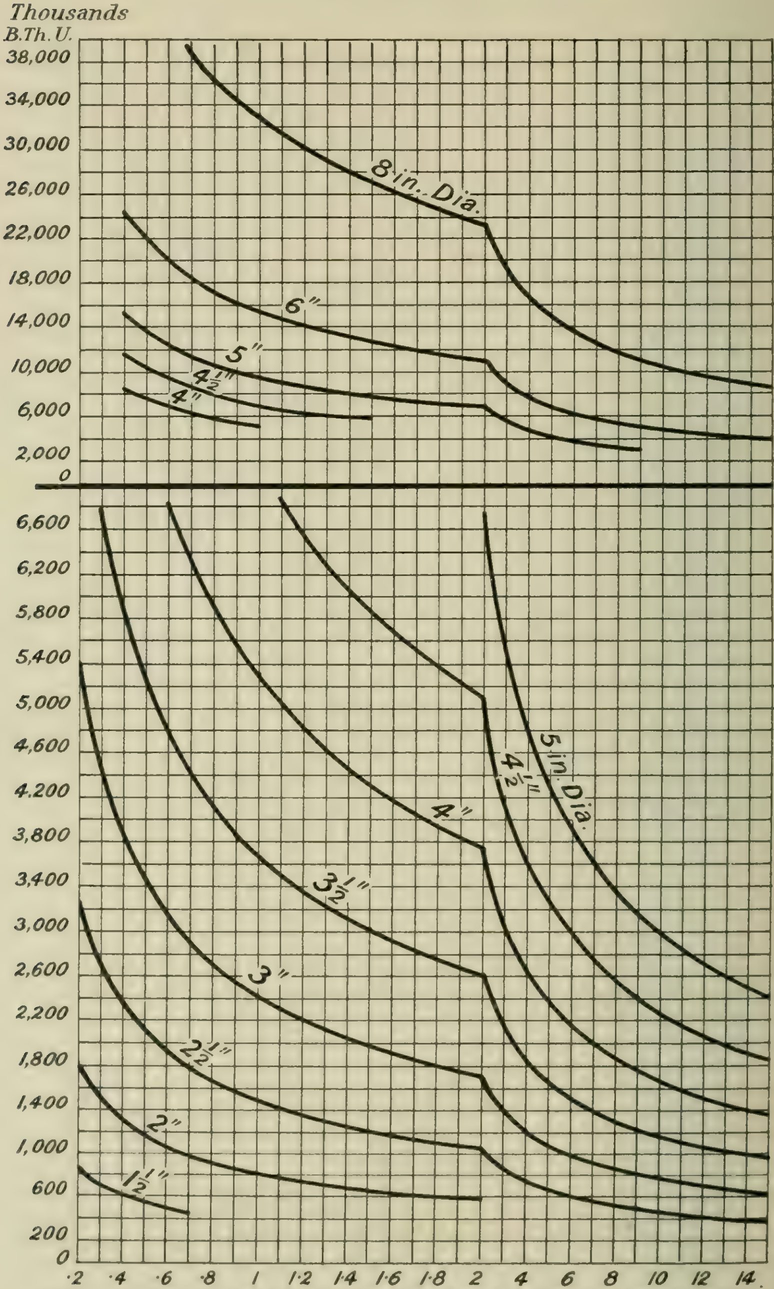
Table XIX. has been prepared for proportional pressure drops per 100 feet run of pipe. For "One Pipe" gravity systems the sizes may either be picked out from Table XIX. or from the charts, but the next higher size to the one obtained should be chosen on account of the same piping conveying the condensation as well as the steam.

Chart 21.—Steam pressure 5 lb. per square inch. Capacity of circuit in B.Th.U. per hour.



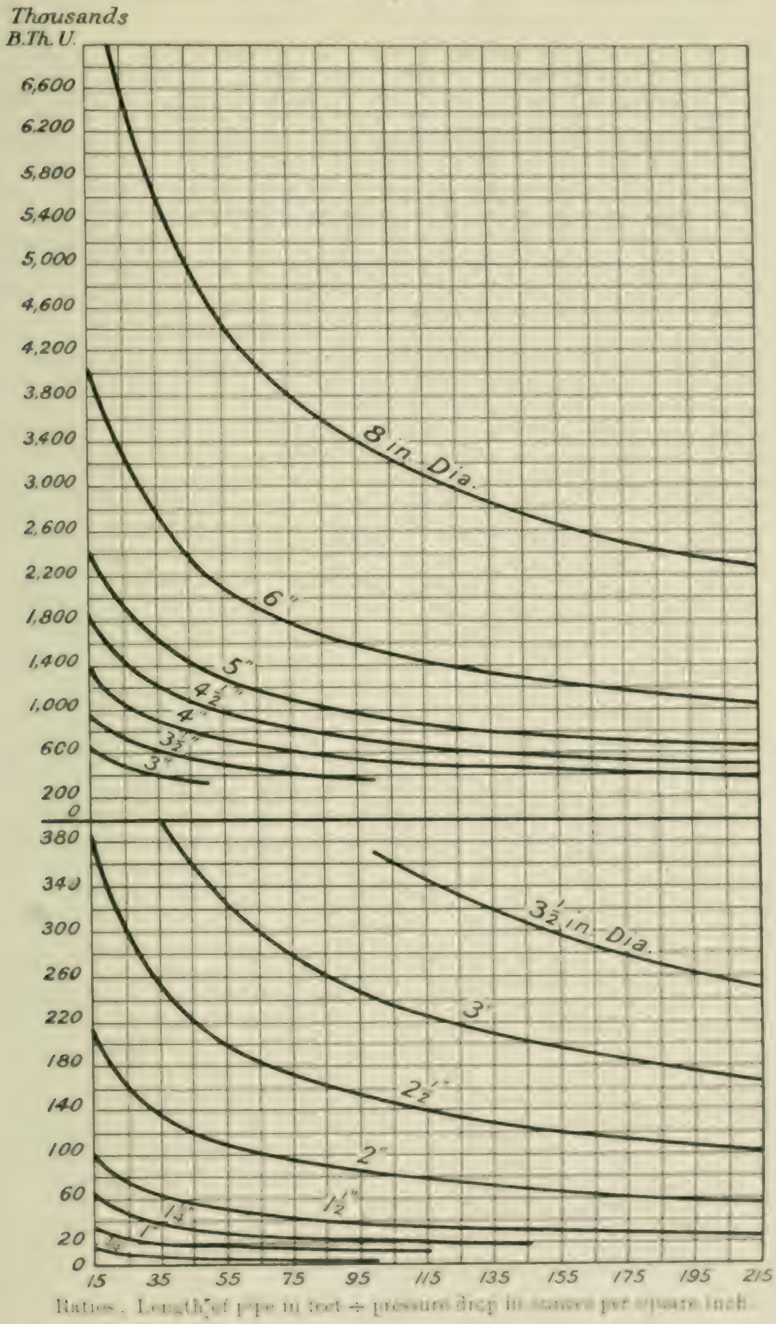
Ratios : Length of pipe in feet ÷ pressure drop in ounces per square inch

Chart 22.—Steam pressure 5 lb. per square inch. Capacity of circuits in B.Th.U. per hour.



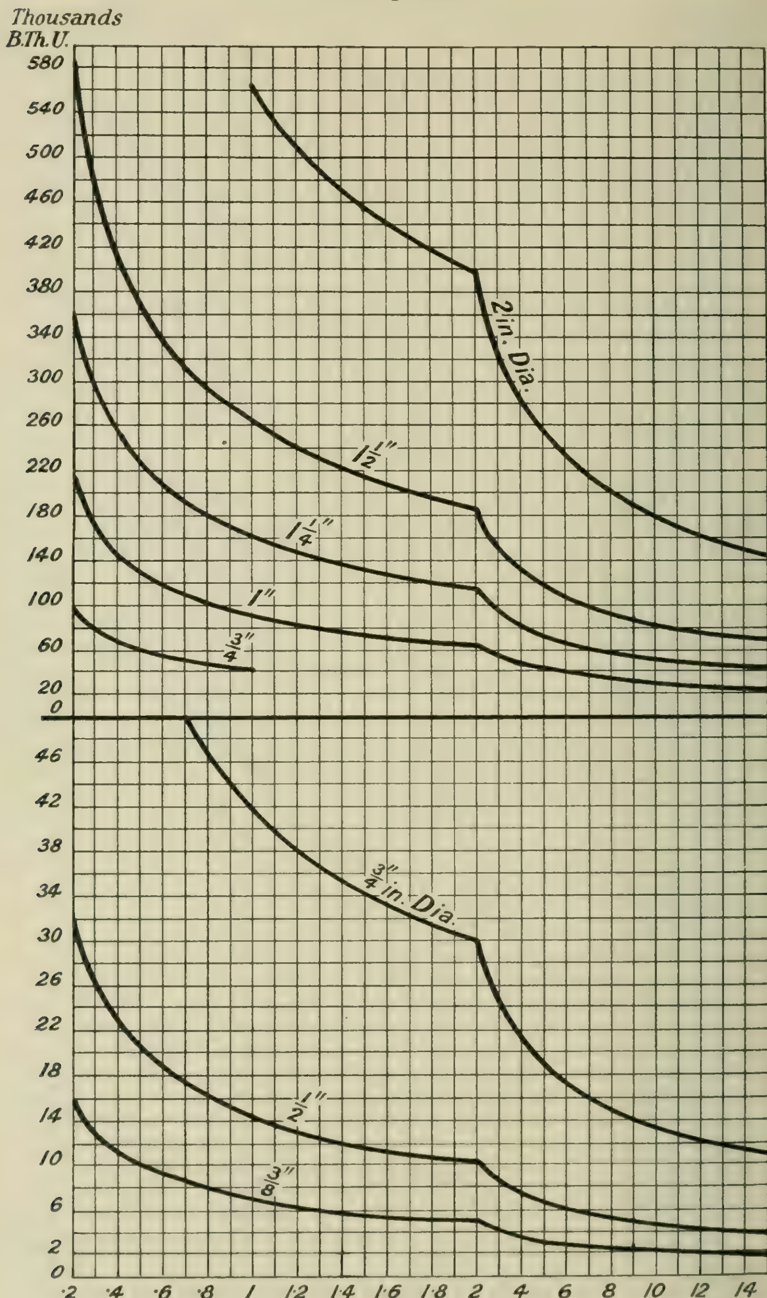
Ratios; Length of pipe in feet ÷ pressure drop in ounces per square inch.

CHART 23.—Steam pressure 5 lb. per square inch. Capacity of circuits in B.Th.U. per hour.



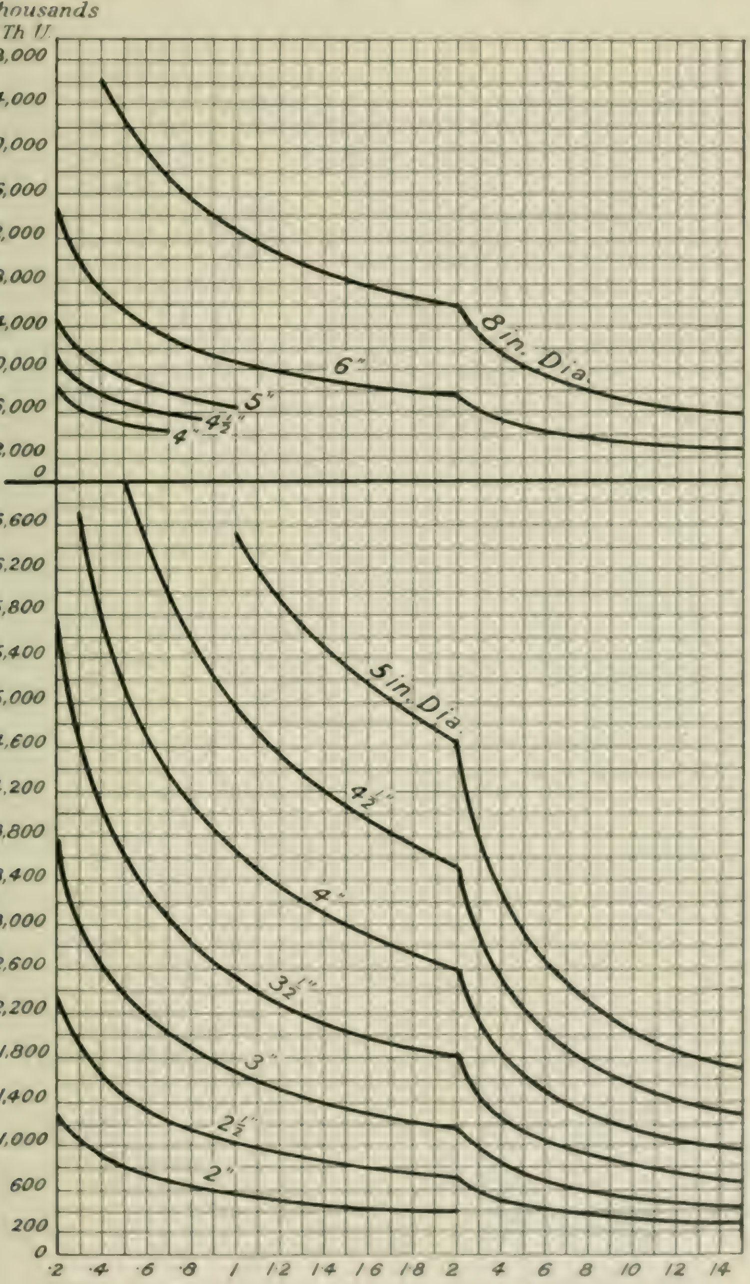
Ratios, Length of pipe in feet ÷ pressure drop in ounces per square inch.

Chart 24.—Steam pressure 1 lb. per square inch. Capacity of circuits in B.Th.U. per hour.



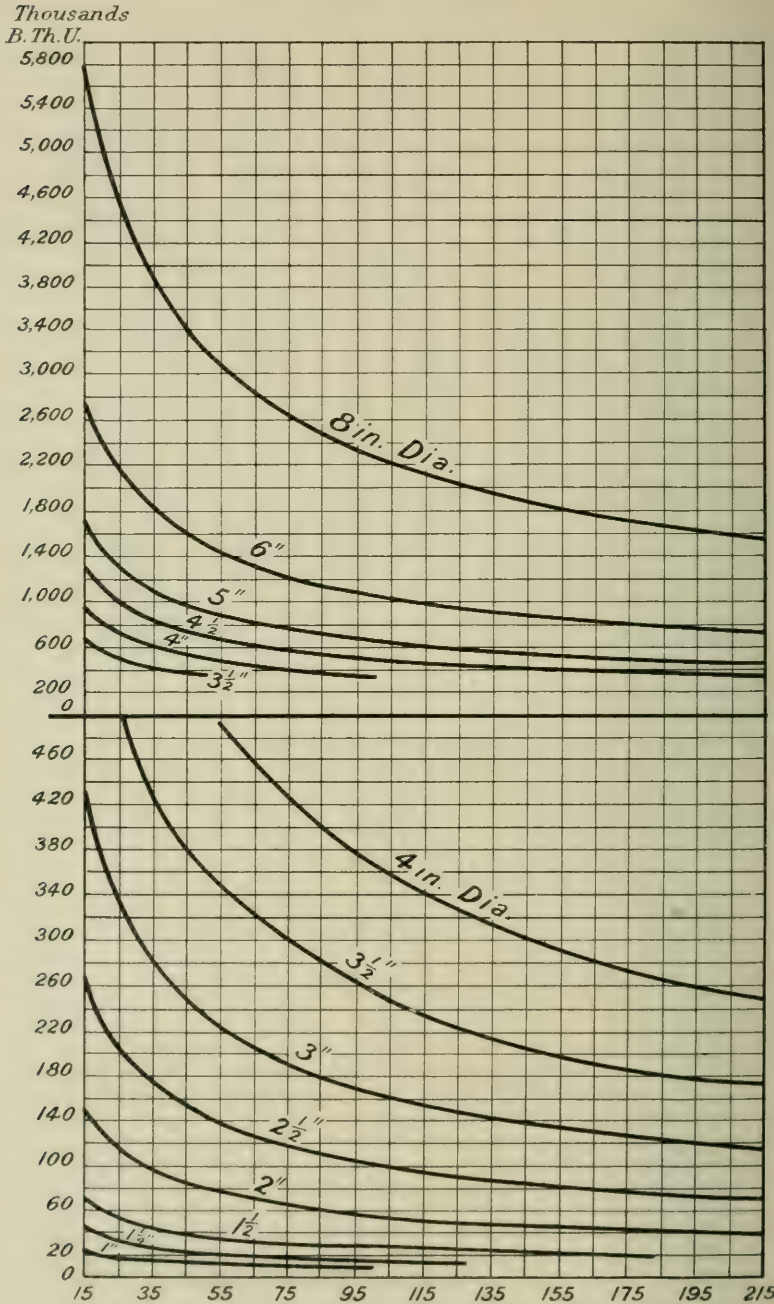
Ratios : Length of pipe in feet ÷ pressure drop in inches of water.

CHART 25.—Steam pressure 1 lb. per square inch. Capacity of circuits in B.Th.U. per hour.



Ratios: Length of pipe in feet ÷ pressure drop in inches of water.

CHART 26.—Steam pressure 1 lb. per square inch. Capacity of circuit in B.Th.U. per hour.



Ratios: Length of pipe in feet ÷ pressure drop in inches of water.

TABLE XIX.

Two-Pipe Gravity Steam Apparatus. Capacity of Pipes in British Thermal Units and in Square Feet of Heating Surface. Each square foot is assumed to transmit 250 B.Th.U. per hour.

When applying to One-Pipe Systems, increase the diameters obtained by one size.

Diameter of pipe in Inches.	Proportional pressure drop in ozs. per square inch per 100 feet of pipe.*						
	1	1½	2	2½	3	4	
	Capacity of pipes.						
¾	—	—	8,600 35	9,600 39	10,500 42	12,100 49	B.Th.U. Sq. ft.
1	13,000 52	16,000 64	18,400 74	20,600 83	22,600 91	26,200 105	B.Th.U. Sq. ft.
1¼	23,200 93	28,500 114	32,800 131	36,800 147	40,200 161	46,400 186	B.Th.U. Sq. ft.
1½	37,800 151	46,200 185	53,500 214	59,800 239	65,500 262	75,600 302	B.Th.U. Sq. ft.
2	81,200 325	100,000 400	115,000 460	128,000 512	140,000 560	163,000 652	B.Th.U. Sq. ft.
2½	148,000 592	181,000 724	210,000 840	234,000 936	257,000 1,028	296,000 1,184	B.Th.U. Sq. ft.
3	240,000 960	295,000 1,180	340,000 1,360	381,000 1,524	418,000 1,672	480,000 1,920	B.Th.U. Sq. ft.
3½	368,000 1,472	450,000 1,800	523,000 2,092	582,000 2,328	640,000 2,560	737,000 2,948	B.Th.U. Sq. ft.
4	528,000 2,112	647,000 2,588	748,000 2,992	835,000 3,340	915,000 3,660	1,056,000 4,224	B.Th.U. Sq. ft.
4½	718,000 2,872	888,000 3,552	1,015,000 4,060	1,135,000 4,540	1,240,000 4,960	1,436,000 5,744	B.Th.U. Sq. ft.
5	946,000 3,784	1,160,000 4,640	1,340,000 5,360	1,500,000 6,000	1,640,000 6,560	1,892,000 7,568	B.Th.U. Sq. ft.
6	1,550,000 6,200	1,900,000 7,600	2,200,000 8,800	2,450,000 9,800	2,680,000 10,720	3,100,000 12,400	B.Th.U. Sq. ft.
1	2	3	4	5	6	7	

* To obtain pressure drop in inches of water, multiply ounces per square inch by 1·73.

Sizes of Radiator Valves.—These may be obtained either by the aid of the Steam Charts or from Table XIX. In either case, only a low fall of the steam pressure should be allowed. Take the next higher size obtained in this way when the valves are for "One-Pipe" systems.

Application of Steam Charts.—Where only a very approximate method of sizing pipes is sufficient, all that is necessary is to proportion roughly the available pressure drop between the different sections of piping. This method is often sufficiently accurate for ordinary gravity systems. On the other hand,

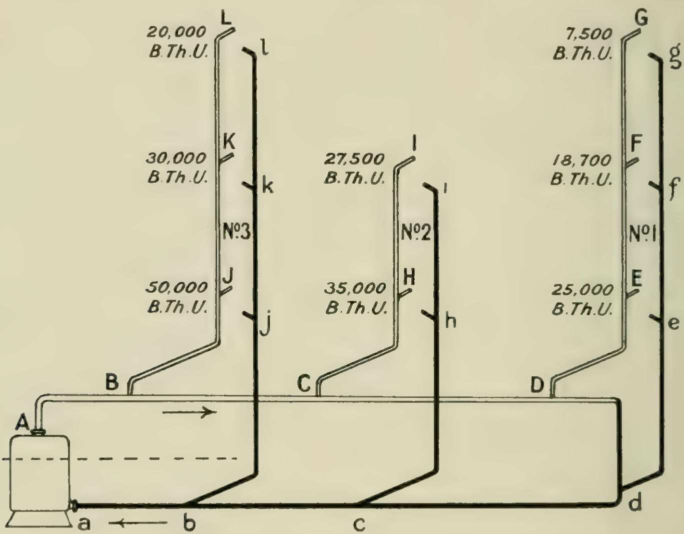


FIG. 150.—Two-pipe "up-feed" system.

where the whole of the initial steam pressure should be absorbed in pipe friction, the process of calculating can be extended by ascertaining the exact pressure that may be expended upon each section of the piping. The more complete and refined process is suitable for atmospheric and vacuum installations. The charts may be used with simple adjustment for steam pressures between that of the atmosphere and a gauge pressure of 5 lb. per square inch.

Fig. 150 and the tabulated particulars will show how the piping may be sized for a "two-pipe" ordinary gravity system,

where the initial steam pressure is 5 lb. per square inch, and where the drop of pressure between the boiler and the top of the end riser is limited to about 4 oz. per square inch.

(1) After the general design has been prepared, arrange into sections, and determine the thermal capacity of each.

(2) Ascertain the length of piping for each section, and to each add the equivalent for fittings. See Table XXIII. in appendix.

(3) Proportion the drop of pressure between the sections of the mains. The best allowance for each is gleaned when the process of sizing is commenced, but the total stipulated should not be exceeded.

(4) Divide the total length of each section by the pressure drop allowed, and from the ratio and thermal capacity pick out the diameter from the charts.

The sizes of the returns may be added from Table XVIII., p. 273.

Particulars of Fig. 150. Steam Pressure 5 lb. per square inch, and Approximate Drop of Pressure 4 oz. per square inch.

Item	Section.	Capacity in B.Th.U.	Net length in feet. +	Equiv. for fittings. =	Total length l	Approx. pressure drop in ozs. per sq. in. p.	Length ÷ pressure drop $\frac{l}{p}$	Diameter of steam pipe d.	Diameter of corresponding returns d_r.
Main	AB	213,700	30	18	48	1	48	2½	2
,,	BC	113,700	40	6	46	1	46	2	1½
,,	CDE	51,200	75	15	90	2	45	1¼	1¼
Riser 1	EF	26,200	12	2	14	0·25	56	1¼	1
,,	FG	7,500	12	10	22	0·25	88	1	¾
Branch	E	25,000	4	6	10	0·5	20	1	¾
,,	F	18,700	4	6	10	0·25	40	1	¾
Riser 2	CH	62,500	14	12	26	0·75	35	1¼	1¼
,,	HI	27,500	12	10	22	0·5	44	1¼	1
Branch	H	35,000	6	6	12	0·75	16	1¼	1
Riser 3	BJ	100,000	6	14	20	0·5	40	2	1½
,,	JK	50,000	12	2	14	0·5	28	1¼	1
,,	KL	20,000	12	10	22	0·5	44	1	¾
Branch	J	50,000	4	6	10	1·0	10	1¼	1
,,	K	30,000	4	6	10	0·75	13	1	¾
1	2	3	4	5	6	7	8	9	10

It will be observed in the tabulated matter that the total pressure drop for the steam risers differs, and this plan may be followed in practice, owing to the smaller pressure drop in the first portion of the steam mains.

In Fig. 151, a portion of an atmospheric system is given. Here the initial boiler pressure is taken as equal to a column of 24 inches of water, whilst it is assumed that only about

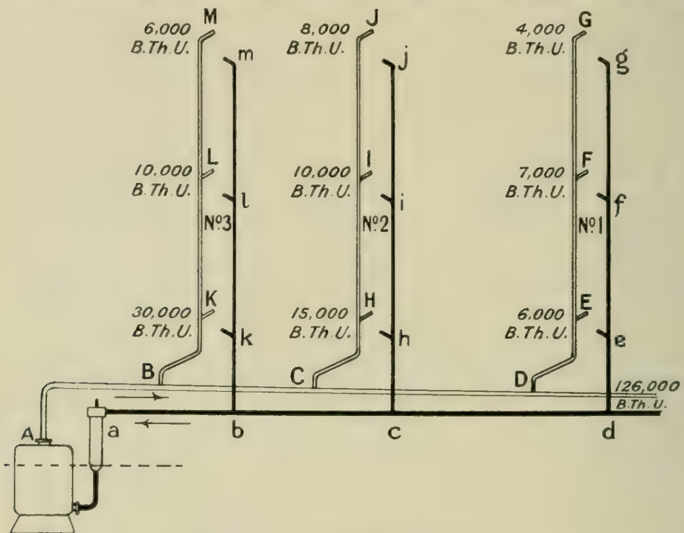


FIG. 151.—Atmospheric system.

10 inches of pressure may be absorbed by the piping to the point D. The procedure to be followed at the outset resembles that in connection with the previous problem, viz. to divide into sections, and to determine the thermal capacity and the total length of each. This being accomplished, proportion approximately the 10 inches of pressure between the three sections of the main, and pick out the diameters from the charts.

The pressure that should be absorbed by each riser is obtained by deducting from the initial or boiler pressure that absorbed by the main piping to the points of junction. For example, if the approximate pressure heads utilized by the sections AB, BC, and CD are 3, 2, and 5 inches of water, then the pressures at the points B, C, and D would be assumed as 21, 19, and 14 inches respectively. The latter values represent

the pressures that would be available for sizing the risers, and these may be divided between the different sections in any proportion that is found suitable.

In a similar manner, the pressure at the riser branches is obtained for determining their size.

When the trial diameters have been obtained, the actual pressure they absorb may be ascertained by taking the ratios agreeing with these diameters and their thermal capacities, and by dividing the total length of the sections by the ratios in question. During this process, it will be seen whether any alteration in the sizes of the pipes should be made or not, by noting the difference which exists between the total pressure absorbed by the pipes and that actually at disposal.

Particulars of Fig. 151.

Initial Steam Pressure = 24 Inches Water. Pressure that may be absorbed to Point D approximately 10 inches.

Item.	Section.	Capacity in B.Th.U.	Net length in feet. +	Equivalent for fittings.	Total length in feet l.	Trial press. drop in inches h_1.	Length ÷ press. drop $\frac{l}{h_1}$	Trial dia. ins. d_1.	Pressure available at each point in ins. of water.	Ratio from table of capacity and diameter.	Actual press. drop in ins. of water for diameter chosen $h = \frac{l}{\text{ratio}}$	Dia. of steam pipe.	Dia. of return.
Main	AB	222,000	30	20	50	3	17	2½	B 21·62	21	2·38	2½	1¼
,,	BC	176,000	25	8	33	2	17	2½	C 20·68	35	0·94	2½	1¼
,,	CD	143,000	60	8	68	5	14	2	D 16·18	15	4·5	2	1¼
Riser 1	DE	17,000	15	10	25	6	4	¾	—	6·5	3·85	¾	[illegible]
,,	EF	11,000	12	2	14	3	5	¾	—	1·7	8·24	[illegible]	[illegible]
,,	FG	4,000	15	8	23	5	5	[illegible]	—	15·0	1·54	[illegible]	[illegible]
Branch	E	6,000	5	8	13	8	1·6	[illegible]	12·3	1·35	9·0	[illegible]	[illegible]
,,	F	7,000	4	8	12	5	2·4	[illegible]	4·6	4·5	2·66	[illegible]	[illegible]
Riser 2	CH	33,000	12	12	24	7	3·5	1	—	8·0	3·0	1	[illegible]
,,	HI	18,000	12	3	15	5	3·0	¾	—	5·6	2·68	¾	[illegible]
,,	IJ	8,000	15	10	25	7	3·6	[illegible]	—	3·6	6·94	[illegible]*	[illegible]
Branch	H	15,000	4	12	16	12	1·3	[illegible]	17·68	0·95	16·8	[illegible]	[illegible]
,,	I	10,000	4	10	14	7	2·0	[illegible]	15·0	2·3	6·1	[illegible]*	[illegible]
Riser 3	BK	46,000	20	15	37	10	3·7	1	—	4·0	9·25	1	[illegible]
,,	KL	16,000	14	4	18	5	3·6	¾	—	7·0	2·6	¾	[illegible]
,,	LM	6,000	14	10	24	6	4·0	[illegible]	—	6·5	3·7	[illegible]*	[illegible]
Branch	K	30,000	4	16	20	11	2·0	¾	12·37	2·0	10·0	¾	[illegible]
,,	L	10,000	6	12	18	6	3·0	[illegible]	9·77	2·3	7·8	[illegible]	[illegible]
1	2	3	4	5	6	7	8	9	10	11	12	13	14

* The throttling of these pipes will be essential

Perhaps an example in greater detail will aid the explanation given in reference to the more refined method of computation. Take the end Riser No. 1, Fig. 151, where the steam pressure at point D works out at about 16·18 inches of water, and which should be absorbed by the sections DE, EF, and FG. The trial diameter DE which supplies 17,000 B.Th.U. has been taken as $\frac{3}{4}$ inch, and the ratio agreeing with the two values is found in Chart 24 to be about 6·5. Section DE has a total length of 25 feet, and when divided by the ratio 6·5 gives the head absorbed as 3·85 inches of water.

For the section EF, the trial diameter selected was $\frac{3}{4}$ inch, but this will be found rather large, so the next smaller bore is taken. The ratio on Chart 24 agreeing with a thermal capacity of 11,000 B.Th.U. and a $\frac{1}{2}$-inch diameter pipe is 1·7, and this divided into the total length of section gives 8·24 inches as the pressure absorbed.

Section FG has a trial diameter of $\frac{1}{2}$ inch, and for a capacity of 4000 B.Th.U. gives a ratio of 15. The length of 23 feet divided by 15 gives 1·54 inches as the head absorbed by this section.

The total pressure absorbed by the riser is therefore $3·85 + 8·24 + 1·54 = 13·63$ inches, whilst that available at D is shown to be 16·18 inches. Where the difference is rather marked, as in this case, extra resistance should be added by throttling, or by introducing a short length of piping of a smaller bore.

The pressure available at the branch E will be $16·18 - 3·85 = 12·3$ inches, and as the trial diameter will not absorb this, a smaller pipe is chosen. Chart 24 shows that the ratio for a $\frac{3}{8}$-inch pipe when supplying 6000 B.Th.U. is 1·35, and when divided into the length gives the head absorbed as 9·6 inches. There is still a margin of pressure to spare at this branch of $12·3 - 9·6 = 2·9$ inches. Branch F has a diameter of $\frac{1}{2}$ inch, and the ratio for this when supplying 7000 B.Th.U. is about 4·5. Dividing length of branch by the ratio gives 2·66 inches as the head or pressure absorbed. Here the available pressure at F is $16·18 - 3·85 - 8·24 = 4$ inches, which is a little in excess of that absorbed by friction by the branch.

CHAPTER XXIII

BOILERS

THE fuels used in heating apparatus may take either the solid, the gaseous or the liquid form, and although the former is the cheapest, the others possess advantages under special conditions. Moreover, the hot waste gases from internal combustion engines could be utilized to a greater extent than at present by providing suitable elements for the absorption of heat in their passage from the engines. A supplementary heater would be essential when the engines were not running, or where the waste gases did not yield sufficient heat, but this case is only similar in principle to that of exhaust steam heating where a connection is made with a "live" steam pipe to make good any deficiency of the "exhaust."

Boilers may be broadly classified as "high" and "low" pressure types, and both take a large variety of forms, but it is chiefly with the latter class that the writer proposes to deal.

The materials used in the construction of boilers are wrought iron, mild steel, cast iron, and copper. The latter, however, is not used so much for boilers for warming buildings, as for heating water for domestic and other purposes. Copper boilers are necessary where the water has a corrosive action upon iron, or where the discoloration of the water is undesirable. When well made, copper boilers are very durable, provided that they are subjected to proper use, but it is no uncommon thing to find weak points in the brazing work. At the present time, where moderately large boilers are required, the cast-iron sectional types are generally used. These are advantageous in that they offer facilities for great variation in design, are convenient for handling and installing where space

is limited, and as a rule are less affected by corrosion than wrought iron and steel ones. For small boilers, or where high pressures require to be carried, wrought iron and steel often enter into their construction.

General Aspects of Heating Boilers.—These are operated very differently from power boilers, the firing periods being at intervals more or less prolonged, whilst in power boilers, the firing process is more regularly and intelligently performed. The successful action of a heating boiler depends upon a number of factors all of which are correlated. These are principally as follows:—

(1) The efficiency of the boiler surfaces for the transmission of heat.
(2) Rate of firing.
(3) Quality of fuel consumed.
(4) Chimney draught.
(5) Ratio of boiler surface to grate area.
(6) Freedom with which water circulates through and from a boiler.
(7) Suitability of fire grate.
(8) Size of fire pot.
(9) The time the boiler is working on one charge of fuel.

Efficiency of Boiler Surfaces.—The quantity of heat that will pass through any portion of a boiler surface in unit time depends upon its relative position to the fire, the temperature difference of its two sides, the velocity with which the heated products of combustion sweep over it, the velocity of circulation, and its degree of cleanliness. Direct surfaces, or those facing the fire, receive radiant heat as well as that from the products of combustion. On the other hand, indirect surfaces, or those forming the flues, receive their heat mainly from the gases passing over or impinging against them. For these reasons, a heating boiler should contain as much direct surface as possible, and especially is this the case where the rate of firing is very low.

The precise value of the radiant heat cannot be ascertained very well, as it varies considerably at different stages of a charge of fuel, and according to the conditions under which the firing operations are performed. In any case, its value is relatively

high. Direct surfaces are also easy to keep clean, and this is of great importance in the case of boilers that are likely to receive indifferent attention.

Indirect boiler surfaces often have their value overstated, it being assumed that they cool down the products of combustion to a relatively low temperature. Instead of effecting this in the way desired, long tortuous flues often have the effect of producing a sluggish chimney draught, whilst the soot deposited upon their surfaces retards the transference of heat. That these facts have been grasped by some of the best boiler makers, is shown by the simpler designs they now produce when compared with those of a few years ago.

Rate of Firing.—The rate of firing has a direct influence upon the temperature in the furnace and upon the gases passing to the chimney, and it also affects the completeness or otherwise of the combustion. The quantity of fuel consumed per square foot of grate per hour varies from 2 to 12 lb., but in the case of small boilers, it does not as a rule exceed 6 lb.

When a fuel is properly consumed, its contained carbon is converted into carbonic acid gas, and each pound of carbon yields 14,500 B.Th.U. On the other hand, if combustion is incomplete, carbon monoxide is formed, and the heat given up is only about 4400 B.Th.U. per pound of carbon burned. With very slow rates of combustion, and especially when soft bituminous fuels are used, the efficiency of a boiler rapidly falls, for the temperature of the furnace is relatively low, and for a period, combustion is incomplete.

The temperature of combustion varies widely in heating boilers during the burning of one charge of fuel, and as a rule the gases in the furnace will not exceed greatly at any time 1500° F., whilst they may fall considerably below 800° F. when a fresh charge of fuel is added.

Quality of Fuel.—Since heating boilers are assumed to require little attention, anthracite coal and coke should be the principal fuels used. Bituminous coals make too much smoke, and rapidly soot up the flues. Both anthracite coal and coke consist principally of carbon, and are relatively free from volatile matter.

The calorific value of fuels varies considerably, the following table giving a few averages:—

TABLE XX.

Fuel.	B.Th.U. per lb.	Fuel.	B.Th.U. per lb.
Coal, anthracite	15,000	Coke (gas) . .	11,000
,, Welsh medium . .	13,900	Wood . . .	7,500
,, Newcastle	14,000	Peat	7,000
,, Scotch	13,500	Petroleum . .	20,000
Coke	13,000		

Chimney Draught.—For a boiler to operate satisfactorily, it is imperative that a good draught be obtainable, for unless the requisite quantity of air can be passed through the fire, the combustion will be imperfect, and insufficient fuel will be consumed. The design of a boiler may be modified to some extent where a liberal draught is available in order that the gases may sweep over and against its heat-absorbing surfaces. High gas speeds increase the efficiency of the surfaces by the displacement of the chilled stratum of air that apparently clings about them.

Ratio of Boiler Surface to Grate Area.—There is a wide difference in heating boilers between the relationship of their grate areas and their total surfaces for absorbing heat. When the boiler surface is expressed as a ratio of the grate area, it varies from about 7 to 1 in small boilers to 30 to 1 and over in some of the larger cast-iron sectional types. The principal effects of these ratios are: (1) The cooling of the products of combustion to a more or less extent; and (2) the increasing or diminishing of the frictional resistance of the flues. It has already been indicated that the effective area of indirect surface is soon passed, owing to the soot it gathers. On the other hand, a very small ratio may be associated with high chimney temperatures, although the rate of combustion is another factor that would require to be taken into account. In general practice, if a well-proportioned boiler is operating with a good draught at its normal rate of combustion, and is kept in a suitable state, the temperature of the gases escaping to the chimney should not greatly exceed 370° F. In unfavourable

cases, the gases may escape at the chimney at a temperature exceeding 600° F., when probably 25 per cent. of the total heat of the fuel passes directly in this manner to waste.

Freedom of Circulation through Boilers.—An unsatisfactory circulation may arise through defective design, or a faulty piping arrangement. It is important in hot-water boilers that the fluid should readily flow to the point of escape, otherwise local currents may be set up which raise the water temperature unnecessarily high, and in turn diminish the flow of heat through the surfaces.

If a system of piping is faulty in that the circulation of water through it is too slow, overheating will readily occur, and with certain forms of connections, the water from a boiler may be dislodged. If a piping system is at fault, it is usually revealed by the large temperature difference between the water when leaving and re-entering a boiler.

In steam boilers, it is very desirable that adequate provision should be made for the free circulation of the water and the steam, or "priming" troubles may be caused. The same action may occur in badly proportioned heaters where the water surface is too small to permit of the free disengagement of the steam, or where the steam space is of a too restricted capacity.

Fire Grates.—The grates for heaters take different forms, according to the size of fuel that is used and to its behaviour during consumption. For a fuel such as anthracite coal, the spaces between the fire bars should be narrow, as it breaks up into small pieces in burning. Narrow spaces are also essential for slack or dross, whilst the bars are spaced further apart where coke and bituminous fuels of a moderate size are used. The percentage area of the free air space to the total area of grate varies roughly from 25 to 50 per cent.

To prevent the clinkering of fires, rocking or shaking grates are often provided, but these are more liable to derangement and more expensive to renew than the simple forms of fixed bars.

It will be found occasionally that some of the fire grates are far too large; these are defective in that they do not permit of suitable depths of fuel being used, and the fires are more

difficult to control. Moreover, such grates often permit excessive volumes of air to enter and cool the furnaces, through their not being properly covered with fuel.

Size of Fire Pot.—The depth of the fire pot not only requires to be sufficient to hold a charge of fuel for a given time, but additional space is necessary for the incandescent fuel and ash upon the grate. A further depth of a few inches also requires to be provided between the top of the fuel and the overhanging surfaces. This space permits of the better combustion of the gases at the surface of the fuel, an unnecessary chilling effect through the lower temperature of the boiler surfaces being thus avoided.

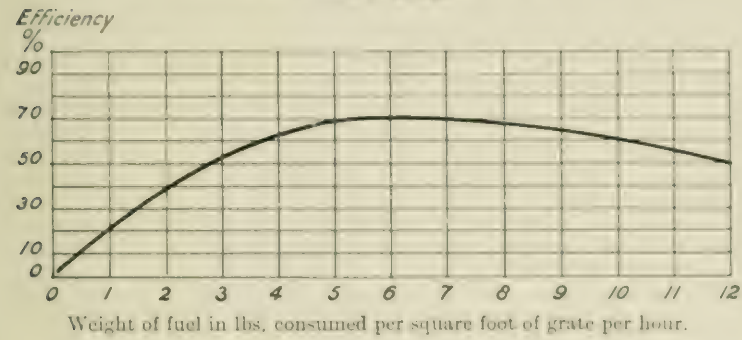
Firing Interval.—The length of time a boiler runs on one charge of fuel directly affects its efficiency, owing to the lowering or the raising of the average temperature of combustion. The interval between two charges will necessarily vary with the size of a boiler unit and the amount of attention it can receive. Boilers of a large size do not as a rule run for more than four hours on one charge, whilst smaller ones often go from eight to ten hours at a time.

Efficiency of Heating Boilers.—From the foregoing, it is clear that if a boiler is said to have a certain efficiency, the circumstances under which it can be obtained should also be given. It is commonly assumed that these boilers have an efficiency of 70 per cent. and over when working under normal conditions. In the writer's opinion, such values are only obtainable with careful operation and with boilers of good design. It will be found that many types, when operated in the usual manner, do not give an average efficiency of 40 per cent.

Charts 27 and 28 show curves of imaginary tests of two different boilers. Chart 27 gives the efficiency curve for different rates of firing when the best performance is obtained on 6 lb. of fuel per square foot of grate per hour. Chart 28 is for a boiler operated in the usual manner, where the interval between the charges varies from 2 to 8 hours. The maximum efficiency for a 2-hour interval is taken as 75 per cent., and for an 8-hour run at 35 per cent. These are average values, for the efficiency would fluctuate over the whole firing interval.

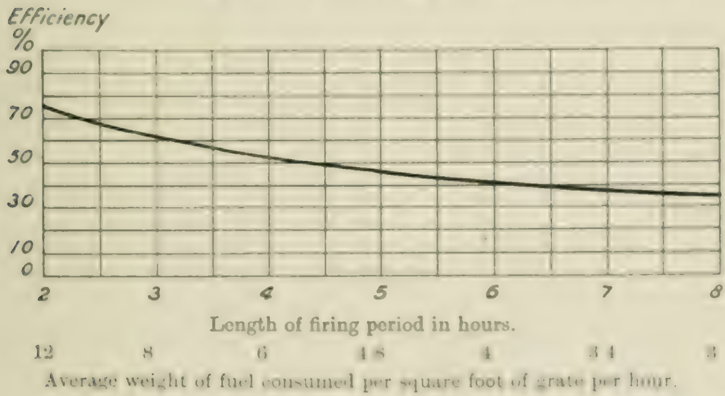
Different designs of boilers when run for varying periods on one charge of a standard fuel, and with suitable draughts,

CHART 27.—Assumed efficiency curve of a heating boiler for different rates of firing.



will give their own characteristic curves. The chief purpose of Charts 27 and 28 is not to give the actual performance of any particular make of boiler, but to show what information is

CHART 28.—Assumed efficiency curve of a heating boiler when run for varying periods on one charge of fuel.



necessary before the most economical type can be selected for any particular case.

Rating of Boilers.—This aspect of boilers has been the subject of much controversy, and from time to time, various suggestions

have been made to establish some common rating basis. At the present, the capacities of boilers are listed in different ways. One method gives the capacity in B.Th.U. per hour without any reference to the conditions that are requisite for obtaining it. Another gives so many feet run of a certain size of pipe, or of the radiator surface that may be supplied. The latter is more unsatisfactory than the first, as the value of the surfaces varies so much under different conditions.

A more rational method is for a maker to guarantee a given efficiency when the boiler is operating under a certain set of conditions which may be reproduced in practice. It would also be an advantage to be supplied with efficiency curves as indicated by Chart 28, for by such curves a comparison could be readily made of the performances of different boilers. In addition to the efficiencies of boilers, the principal details of the tests should be scheduled, such as—

(*a*) Kind of fuel, its calorific value, and size of pieces.

(*b*) Average weight of fuel consumed per square foot of grate per hour.

(*c*) Depth of fuel at the commencement of test.

(*d*) Grate area.

(*e*) Percentage area of free air space in grate.

(*f*) Interval in hours between two charges of fuel.

(*g*) Chimney draught in inches of water.

A knowledge of the size of fuel, its depth, and the free area in grate is necessary in order that the resistance to the entering air in future tests may be similar to that in the original ones.

Size of Boilers.—The easiest way of arriving at the size of a boiler is in terms of its grate area, when its efficiency is given for any set of working conditions. The old method of assuming that each square foot of boiler surface will transmit a certain quantity of heat is defective, as the surfaces cannot be readily measured, even assuming their transmission value were correctly given.

The following formulæ are given with respect to boilers:—

$$U = Awck_c \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

$$R = \frac{Awck_c}{u} \quad . \quad . \quad . \quad . \quad . \quad . \quad (66)$$

$$A = \frac{U}{wek_c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

$$A = \frac{Ru}{wek_c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

where U = capacity of boiler in British Thermal Units per hour.

R = total area in square feet of radiator or equivalent surface.

A = grate area of boiler in square feet.

w = weight of fuel consumed per square foot of grate per hour.

e = efficiency of boiler.

k_c = calorific value of fuel (see Table XX.).

u = heat emitted in B.Th.U. per square foot of radiator surface per hour.

Example 33.—Let the size of boiler be found that is capable of supplying 240,000 B.Th.U. per hour where the fuel has a calorific value of 12,500 B.Th.U. per lb., and the rate of firing equal to 8 lb. per square foot of grate. Assume the efficiency of the boiler for the conditions given as 70 per cent.

By Formula 67—

$$A = \frac{U}{wek_c}$$

Substituting values, $\quad A = \dfrac{240{,}000}{8 \times 0{\cdot}7 \times 12{,}500}$

when **A = 3·43 square feet.**

Size of Boiler Units.—In large plants, it is more economical to operate with two boiler units than with one, owing to the great difference between the minimum and maximum demand for heat during the heating season. The best size of unit will depend very largely upon the point from which it is estimated, and upon whether any duplication is considered essential or not. One method where two boiler units are adopted is to make each capable of supplying two-thirds the ordinary maximum demand, this arrangement permitting either the one or the other to be independently operated over a great portion of the heating season. For abnormal demands of heat, both boilers are operated together.

Another method of sizing two units is to make one capable of supplying two-thirds and the other a little less than one-half of the ordinary maximum heat demand. The smaller unit is advantageous for mild weather, whilst these two sizes can be economically operated together when the demand for heat is greatest.

In some cases, the number of boiler units is regulated by the space at disposal, and a battery of four or more may be required. All are joined with common headers, so that any one may be operated independently of the others. A number of small units, however, are better avoided if practicable.

For ordinary gravity steam-heating systems, it is specially desirable that the boiler units should be of ample size, otherwise the water-line will fluctuate very much. Should the condensing capacity of a system exceed the steam supply, there is danger of the water disappearing from the boiler into the return mains, owing to the partial vacuum caused.

Forms of Boilers.—Fig. 152 gives a sectional hot-water boiler of a fairly large size, and the path taken by the products of combustion may be easily followed. To get the best results from this form of boiler, a good chimney draught is necessary.

In Fig. 153, another hot-water boiler is shown, but of a simpler form, the products of combustion having a less distance to traverse than in the case of Fig. 152. There are many styles of cast-iron sectional boilers, but their principal features have been fairly well considered. Moreover, catalogues may be readily obtained from the various manufacturers of these products.

A steam boiler showing the usual mountings is indicated in Fig. 154. This is also a cast-iron sectional one, but it is protected with a non-conducting material which is covered with a thin sheet-iron casing. All independent boilers should be treated in some similar way to reduce the heat loss from them.

Where high-pressures are carried, cylindrical boilers of the "Cornish," "Lancashire," or similar types are frequently used, their shape being the more suitable for resisting the internal pressure. The various "water-tube" and "multitubular" forms of boilers are principally used in conjunction with power plants, and where a high evaporative efficiency is required.

"Down-Draught" Boilers.—Sometimes a "down-draught"

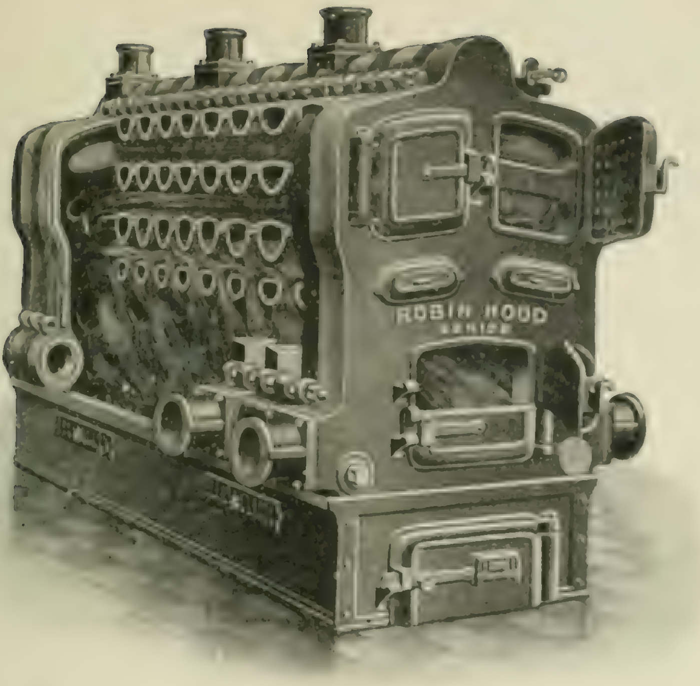


FIG. 152.—Cast-iron sectional boiler. By the Beeston Foundry Company.

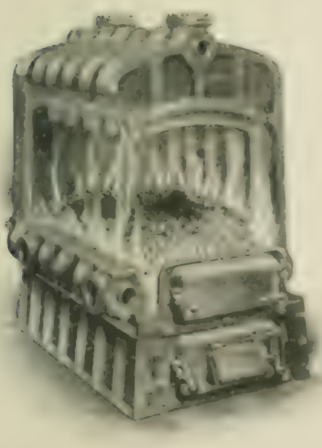


FIG. 153.—Cast-iron sectional boiler. By the National Radiator Company.

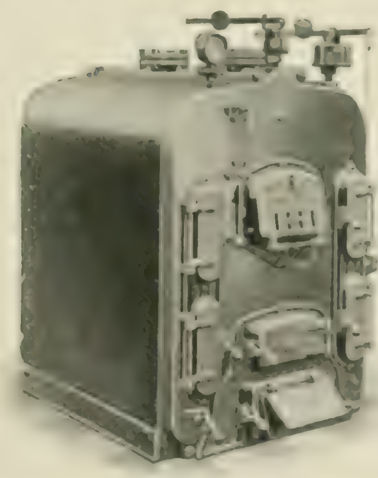


FIG. 154.—Steam boiler showing "trimmings." By the National Radiator Company.

type of low-pressure boiler is used where an attempt is being made to prevent the emission of black smoke. The feature introduced here causes the products of combustion to pass first downwards through the incandescent fuel before passing into the flues and to the chimney. The more satisfactory method of dealing with black smoke, however, is in boilers where mechanical firing can be introduced.

Boiler Mountings.—The common mountings for a steam boiler consist of the following: Pressure gauge, water gauge, relief or safety valve, and automatic draught regulator. In some cases, an automatic water feeder is also installed.

Water Gauge.—It is the usual practice to attach the water gauge of a boiler to a pipe column, whilst to the latter are fitted three "try cocks" for verifying the accuracy of the gauge. The "try cocks" are located one above another, the position of the centre one coinciding with the normal water-line. Water columns are sometimes provided with alarms to indicate when the water-level is either too high or too low.

Automatic Water Feeders for steam boilers usually operate with float valves. These appliances are located so that their water-levels, when the valves are closed, coincide with the boiler water-line. This method of renewing water is only suitable where small volumes are concerned, and in order to give satisfaction, the float valves require to be well made, whilst the water-level in the feeders should be kept as steady as possible.

Safety Valves.—For either hot water or steam boilers, "dead-weight" or "spring" safety valves are suitable. For the latter type, however, the springs used should be made of a non-corrosive material. The safety valves on these boilers are rarely tested, so it is imperative that a pattern should be adopted that is not liable to stick or to be uncertain in its action.

Boiler Chimneys.

It has been shown that the successful operation of a boiler depends largely upon an adequate draught, for unless the requisite quantity of fuel can be properly consumed, the full capacity of the boiler cannot be realized.

Chimney Draught.—The draught of a chimney is produced by the differential pressure of the atmosphere and the heated gases in the chimney, the intensity of the draught being directly proportional to the square root of the temperature difference and to the height of the chimney.

The available pressure that creates the draught is absorbed in different ways, such as by—

(*a*) The resistance of the fire grate and the channels conveying the air to that point.

(*b*) The thickness of the fire and the kind and size of fuel.

(*c*) The resistance offered by the boiler flues.

(*d*) The resistance of the chimney itself.

The proportions in which the pressure is utilized will necessarily vary with different boilers, and even at different periods with any one type. From a general standpoint, the best that can be done is to make a chimney sufficiently large, and to effect exact regulation by the aid of dampers.

Different results will also be obtained with chimneys of different construction, and according to the manner in which they are protected. For example, iron-pipe chimneys, when exposed to the elements cool down the products of combustion, and the draught suffers in consequence. Leaky chimneys have a similar effect.

In any flue or chimney, the velocity of the products of combustion will not be uniform over its whole cross-sectional area, so that in order to estimate their average speed the velocity pressure should be taken at a number of points over the whole cross-sectional area.

Where the velocity pressure is obtained by a suitable water gauge, the speed of the chimney gases may be obtained by the following rule:—

$$v = 18{\cdot}3\sqrt{\frac{h}{D_a}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

where v = velocity in feet per second.

h = velocity pressure in inches of water.

D_a = density of the products of combustion per cubic foot. This is usually taken as the equivalent of air at the same temperature.

For the density of air at different temperatures see Table XXII. of Appendix.

Example 34.—At what speed do the products of combustion flow through a flue or chimney when the water gauge records 0·15 inch? Temperature of gases, 400° F.

By Formula 69—

$$v = 18 \cdot 3\sqrt{\frac{h}{D_a}}$$

The density of the products of combustion may be taken as 0·0461 lb. per cubic foot.

Substituting values, $v = 18 \cdot 3\sqrt{\frac{0 \cdot 15}{0 \cdot 0461}}$

when v = **33 feet per second.**

Size of Chimneys.—For estimating the cross-sectional area of a chimney, Formula 70 is given. In this, the chimney gases have been assumed at a temperature of 350° F., the external air at 60° F., and the weight of air required per lb. of fuel as 20 lb. The latter value is equivalent to 410 cubic feet of air at 350° F. A liberal margin has been allowed for the various resistances.

$$A = \frac{0 \cdot 003U}{\sqrt{H}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

where A = sectional area of chimney opening in inches.
U = thermal capacity of plant in B.Th.U.
H = height of chimney in feet.

Example 35.—Determine the cross-sectional area of a chimney 36 feet high for a boiler that has a thermal capacity of 160,000 B.Th.U.

By Formula 70—

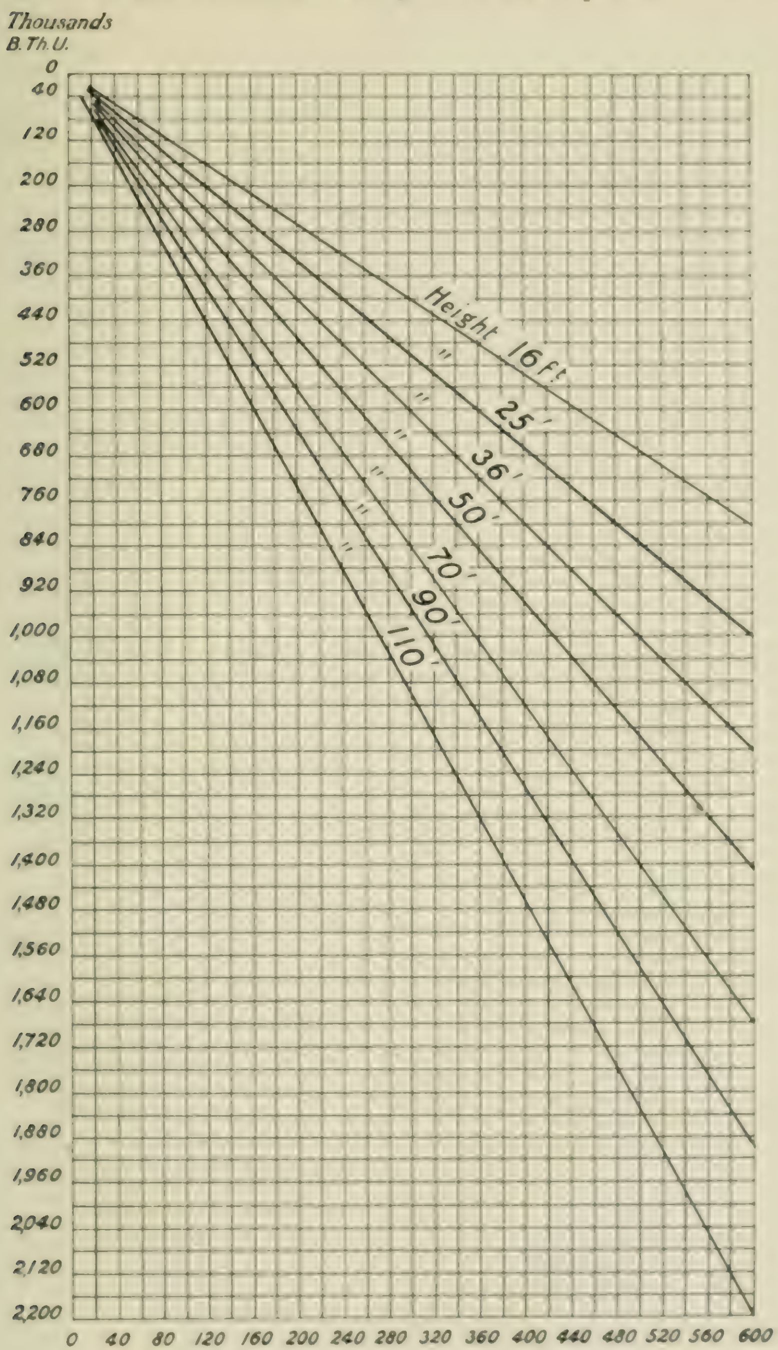
$$A = \frac{0 \cdot 003U}{\sqrt{H}}$$

Substituting values, $A = \frac{0 \cdot 003 \times 160{,}000}{\sqrt{36}}$

when A = **80 square inches.**

Chart 29 has been prepared by the aid of Formula 70. The values are sufficiently liberal, and may be reduced in the majority of cases. No pipe chimney smaller than 6 inches diameter should be used.

CHART 29.—Cross-sectional areas of chimneys for heating boilers where the capacities of the latter are expressed in B.Th.U. per hour.



Cross-sectional area in inches.

CHAPTER XXIV

THE TEMPERATURE CONTROL OF BUILDINGS

In order to effect economy in the operation of a heating plant, some form of temperature regulation is essential owing to the fluctuations of the external atmosphere. It is also desirable to prevent the overheating of buildings in which either physical or mental work is performed, as high temperatures cause the brain to become less alert, whilst a greater expenditure of energy is required to accomplish a particular task.

Temperature regulation may be effected either by hand or by automatic means, the latter being specially suitable for many buildings when a reliable system is installed. Many automatic appliances, however, are very easily deranged, owing either to defective construction or to the introduction of movable parts of far too delicate a nature. If, on the other side, hand control is relied upon, it is often indifferently performed.

For a fairly large heating plant and where a skilled attendant is engaged, hand regulation may be satisfactorily and simply carried out. Under such conditions electric signalling may also be used when the overheating or cooling of a room is recorded automatically at any central point desired.

Automatic Regulation.—There are many systems of temperature control, some being designed for the independent and automatic regulation of the various apartments of buildings, whilst in others the rate of combustion at the boiler is merely governed by locating a thermostat at some central point. The term "Thermostat" is used to designate an appliance that is affected by variations of temperature, and serves either to control the motive power by which the temperature regulation is effected, or supplies from within itself the energy essential

for that purpose. In the construction of thermostats, the principle of differential expansion is largely employed by joining together dissimilar substances in such a way that the one will react upon the other, or work is effected through the vaporization and condensation of very volatile fluids.

The source of energy for the manipulation of the valves where the first class of thermostat is employed may be obtained as follows, by—

(*a*) Some form of mechanism in which springs or weights are used.

(*b*) Compressed air.

(*c*) Partial vacuum.

(*d*) Water pressure.

(*e*) Electricity.

Mechanical Devices that are operated with clock-work gear are not used to any considerable extent on account of the attention required, and, moreover, they are not suitable for plants where the temperature is controlled at a large number of points. They may be used successfully, however, for small systems, but they require to be regularly wound up.

Compressed Air.—This medium is probably the one most largely used in temperature regulation. A supply is readily obtained by a simple form of compressor which may be driven by water by a small electric motor or by any available machinery. The compressors are arranged to be automatic in action, and to maintain an air pressure at the receiver of from 10 to 15 lb. per square inch. Compressed air is readily distributed by means of small tubing to any point desired, whilst any reasonable amount of concentrated power can be easily obtained.

Partial Vacuum.—The use of a partial vacuum for temperature regulation is more suitable where a vacuum system of steam heating is installed than for other forms of heating plants. Under such circumstances, the same pump unit may be frequently used, both as the exhauster for the heating system, and for producing the required degree of vacuum for the temperature regulating equipment. There are cases, however, where this combination would not be economical from an operating standpoint. The general arrangement of the tubing

follows the same plan as for compressed air, but upon reaching a fitting, it is joined to the other side of its diaphragm or piston.

Water Power.—When this is used directly in connection with thermostats, it is suitable for controlling the temperature at the boiler or for regulating the steam supply to a heater. Speaking generally, a hydraulic thermostat should be confined to situations from which the waste water can be readily discharged, or where it is not likely to cause any damage should a leakage occur.

Electricity.—In an electrical equipment, the energy is

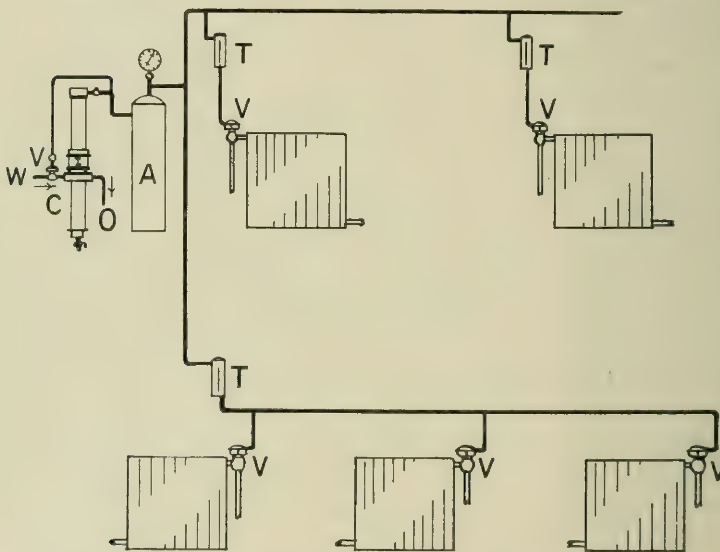


Fig. 155.—Application of thermostats for temperature regulation.

T = thermostats. A = compressed air tank. W = water supply.
V = automatic valves. C = air compressor. O = exhaust water.

derived at a low voltage from a battery, whilst for operating the valves, the circuit is closed or opened by the aid of a mercurial thermometer or by other suitable means. By means of reversible gear, the valves are opened or closed.

Application of Automatic Regulation.—Fig. 155 shows the application of a compressed air system where the radiators on the upper floor are independently controlled, whilst those on the

lower floor are governed by one thermostat. For clearness, the main piping conveying the heating medium is omitted.

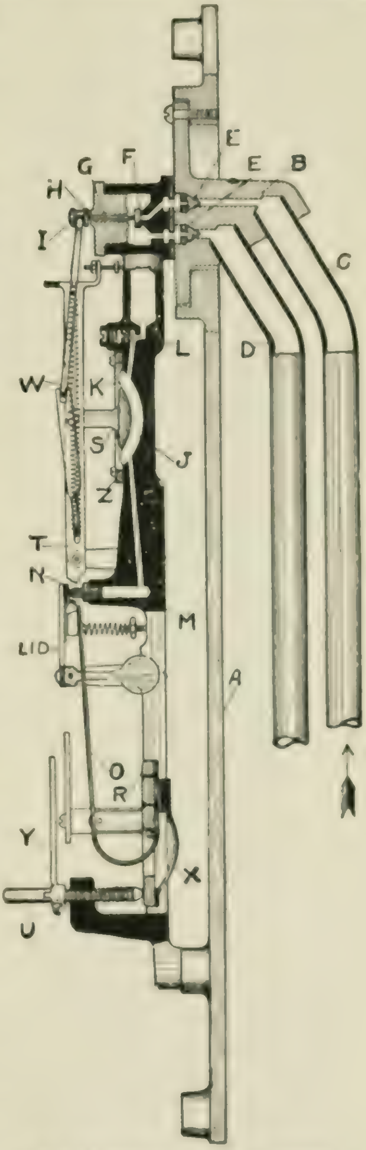


Fig. 156.—Section of "Johnson" thermostat.

Automatic regulation can be applied to any system of hot-water and steam heating, but where a "two-pipe" gravity steam installation is used, the valves on both the inlets and the outlets of the radiators require to be under control. Steam systems, however, are more effectively governed than hot-water plants. Although the automatic valves in Fig. 155 are shown to be directly joined with the radiators, they may be placed in the branch pipes so as to govern the whole of a particular section from one point. The same mode of regulation is extended to indirect heating, and to the operation of dampers, etc., in systems of ventilation. It will be observed that the air from the compressor C is delivered directly into a receiving tank. The storage of compressed air obviates any rapid fluctuation of pressure in the air lines, and so causes the movable parts of the fittings to operate more satisfactorily.

There are various systems of automatic regulation that make use of compressed air, and although the underlying principle is the same in every

case, the fittings used differ considerably in constructional details. Three well-known systems in which compressed air is used are the "Johnson," "Powers," and the "National" Regulator Companies.

A section of the "Johnson" thermostat is given in Fig. 156 and Fig. 157 shows a view of the same appliance with the front cover removed. The pipe C (Fig. 156) joins with the main air

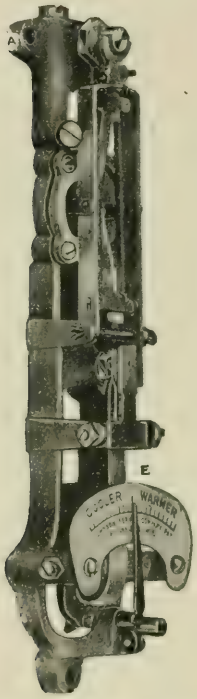


Fig. 157.—Johnson thermostat with front cover removed.

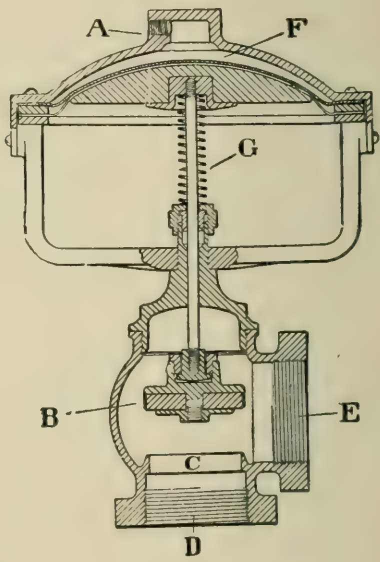


Fig. 158.—Radiator valve.

line and conveys the compressed air to the fitting, whilst D forms the return pipe, or that communicating with the diaphragm chamber of the automatic valve. The passage of the air from the one pipe to the other is controlled by the small valve F, which is operated by the diaphragm K through the agency of compressed air. A compound metallic strip O provides the thermostatic element; its lower end is securely joined with

the appliance, whilst to the upper end is attached a disc to close a small orifice at N. The action of the latter valve, it will be seen, either causes the air pressure to be concentrated upon the diaphragm K, or to effect its release according to the inward or outward movement of the thermostatic element. This element consists of strips of steel and brass soldered together so as to produce movement through their differential expansion and contraction. In Fig. 156 the path taken by the air from C to J is not shown, but a side connection is provided for that purpose. By means of adjustments, the thermostat can be arranged to operate at any desired temperature, whilst the temperature-range between the opening and the closing of the valves may be limited to less than two degrees. This thermostat permits of the leakage of air, but the amount is limited by means of the adjustment screw at L.

A diaphragm radiator valve is given in Fig. 158. It is a direct-acting type, the compressed air pipe joining at A, whilst the spring G raises the valve when the pressure on the diaphragm is relaxed. The air from the diaphragm chamber escapes at the thermostat as soon as the supply is cut off. The diaphragms of these valves should be formed of metal, rubber being of too perishable a nature. In some cases, pistons take the place of the diaphragms to operate the valves.

APPENDIX

TABLE XXI.

Properties of Steam.

These values are taken from Marks and Davis' "Steam Tables," by permission of the authors.

Absolute pressure, lbs. per sq. in.	Gauge pressure, lbs. per sq. in.	Temp., deg. F.	Latent heat, L.	Heat of liquid above 32° F., S.	Total heat of steam above 32° F., H.	Density in lbs. per cubic foot.
1	27·89	101·8	1034·6	69·8	1104·4	0·00300
2	25·86	126·1	1021·0	94·0	1115·0	0·00576
3	23·83	141·5	1012·3	109·4	1121·6	0·00845
4	21·8	153·0	1005·7	120·9	1126·5	0·01107
5	19·77	162·3	1000·3	130·1	1130·5	0·01364
6	17·74	170·1	995·8	137·9	1133·7	0·01616
7	15·71	176·8	991·8	144·7	1136·5	0·01867
8	13·68	182·9	988·2	150·8	1139·0	0·02115
9	11·65	188·3	985·0	156·2	1141·1	0·02361
10	9·62	193·2	982·0	161·1	1143·1	0·02606
11	7·59	197·7	979·2	165·7	1144·9	0·02849
12	5·56	202·0	976·6	169·9	1146·5	0·03090
13	3·53	205·9	974·2	173·8	1148·0	0·0333
14	1·5	209·5	971·9	177·5	1149·4	0·03569
14·7	0·0	212·0	970·4	180·0	1150·4	0·03732
15	0·3	213·0	969·7	181·0	1150·7	0·03806
16	1·3	216·3	967·6	184·4	1152·0	0·04042
17	2·3	219·4	965·6	187·5	1153·1	0·04277
18	3·3	222·4	963·7	190·5	1154·2	0·04512
19	4·3	225·2	961·8	193·4	1155·2	0·04746
20	5·3	228·0	960·0	196·1	1156·2	0·04980
21	6·3	230·6	958·3	198·8	1157·1	0·05213
22	7·3	233·1	956·7	201·3	1158·0	0·05445
23	8·3	235·5	955·1	203·8	1158·8	0·05676
24	9·3	237·8	953·5	206·1	1159·6	0·05907
25	10·3	240·1	952·0	208·4	1160·4	0·0614
26	11·3	242·2	950·6	210·6	1161·2	0·0636
27	12·3	244·4	949·2	212·7	1161·9	0·0659
28	13·3	246·4	947·8	214·8	1162·6	0·0682
29	14·3	248·4	946·4	216·8	1163·2	0·0705
30	15·3	250·3	945·1	218·8	1163·9	0·0728
31	16·3	252·2	943·8	220·7	1164·5	0·0751
32	17·3	254·1	942·5	222·6	1165·1	0·0773
33	18·3	255·8	941·3	224·4	1165·7	0·0795
34	19·3	257·6	940·1	226·2	1166·3	0·0818
35	20·3	259·3	938·9	227·9	1166·8	0·0841
36	21·3	261·0	937·7	229·6	1167·3	0·0863

Vacuum gauge inches of mercury

TABLE XXI—*continued*.

Absolute pressure, lbs. per sq. in.	Gauge pressure, lbs. per sq. in.	Temp., deg. F.	Latent heat, L.	Heat of liquid above 32° F., S.	Total heat of steam above 32° F., H.	Density in lbs. per cubic foot
37	22·3	262·6	936·6	231·8	1167·8	0·0886
38	23·3	264·2	935·5	232·9	1168·4	0·0908
39	24·3	265·8	934·4	234·5	1168·9	0·0931
40	25·3	267·3	933·3	236·1	1169·4	0·0953
45	30·3	274·5	928·2	243·4	1171·6	0·1065
50	35·3	281·0	923·5	250·1	1173·6	0·1175
55	40·3	287·1	919·0	256·3	1175·4	0·1285
60	45·3	292·7	914·9	262·1	1177·0	0·1394
65	50·3	298·0	911·0	267·5	1178·5	0·1503
70	55·3	302·9	907·2	272·6	1179·8	0·1612
75	60·3	307·6	903·7	277·4	1181·1	0·1721
80	65·3	312·0	900·3	282·0	1182·3	0·1829
85	70·3	316·3	897·1	286·3	1183·4	0·1937
90	75·3	320·3	893·9	290·5	1184·4	0·2044
95	80·3	324·1	890·9	294·5	1185·4	0·2151
100	85·3	327·8	880·0	298·3	1186·3	0·2258
110	95·3	334·8	882·5	305·5	1188·0	0·2472
120	105·3	341·3	877·2	312·3	1189·6	0·2683
130	115·3	347·4	872·3	318·6	1191·0	0·2897
140	125·3	353·1	867·6	324·6	1192·2	0·3107
150	135·3	358·5	863·2	330·2	1193·4	0·3320
160	145·3	363·6	858·8	335·6	1194·5	0·3529
170	155·3	368·5	854·7	340·7	1195·4	0·3738
180	165·3	373·1	850·8	345·6	1196·4	0·3948
190	175·3	377·6	846·9	350·4	1197·3	0·4157
200	185·3	381·9	843·2	354·9	1198·1	0·437

TABLE XXII.

WEIGHT OF DRY AIR (ATMOSPHERIC PRESSURE).

Deg. F.	Weight, lbs. per cubic ft.	Deg. F.	Weight, lbs. per cubic ft.	Deg. F.	Weight, lbs. per cubic ft.
0	0·086	140	0·066	375	0·048
10	0·085	150	0·065	400	0·046
20	0·083	160	0·064	450	0·044
30	0·081	170	0·063	500	0·041
40	0·080	180	0·062	550	0·038
50	0·078	190	0·061	600	0·037
60	0·076	200	0·060	650	0·036
70	0·075	212	0·059	700	0·034
80	0·074	230	0·058	750	0·033
90	0·072	250	0·056	800	0·032
100	0·071	275	0·054	850	0·030
110	0·070	300	0·052	900	0·029
120	0·069	325	0·051	950	0·028
130	0·067	350	0·049	1000	0·027

TABLE XXIII.

LENGTH OF PIPING OFFERING THE EQUIVALENT RESISTANCE OF FITTINGS.

Diameter in inches.	Nature of fitting.							
	Sharp elbow or quick curved tee.	Round elbow or diminisher.	Tee.	Return bend.	Radiator connections or square tee.	Angle radiator valve.	Globe valve.	Boiler connections.
½	1·2	0·6	2·0	1·0	2	0·8	2·0	—
¾	1·5	0·8	2·3	1·2	3	1·0	2·3	—
1	2·0	1·0	3·0	1·5	4	1·4	3·0	—
1¼	2·5	1·2	3·5	2·0	5	1·7	3·5	—
1½	3·0	1·5	4·5	2·5	6	2·0	4·5	6
2	4·0	2·0	6·0	3·0	8	3·0	6·0	8
2½	5·0	2·5	7·5	4·0	10	—	—	10
3	6·0	3·0	9·0	5·0	12	—	—	12
3½	7·0	3·5	10·5	5·5	14	—	—	14
4	8·0	4·0	12·0	6·5	16	—	—	16
4½	9·0	4·5	13·5	7·0	18	—	—	18
5	10·5	5·2	15·5	8·0	21	—	—	21
6	13·0	6·5	19·5	10·0	26	—	—	26
7	16·0	8·0	24·0	12·0	32	—	—	32
8	18·0	9·0	27·0	14·0	36	—	—	36
1	2	3	4	5	6	7	8	9

TABLE XXIV.

PROPORTIONAL RESISTANCE OF PIPES OF VARYING SIZES.

Diameter in inches	Diameter of pipes in inches.														
	⅜	½	¾	1	1¼	1½	2	2½	3	3½	4	4½	5	6	8
	Length of pipe in feet which offers the equivalent resistance per foot run of the sizes in Column 1.														
⅜	1	0·223	0·027	0·006	0·0018	0·0007	—	—	—	—	—	—	—	—	—
½	4·47	1	0·122	0·027	0·0081	0·0031	0·00065	—	—	—	—	—	—	—	—
¾	37	8·22	1	0·22	0·067	0·025	0·0054	0·0016	—	—	—	—	—	—	—
1	167	37	4·55	1	0·3	0·115	0·024	0·0076	—	—	—	—	—	—	—
1¼	550	123	15	3·3	1	0·38	0·081	0·025	0·0092	0·0041	0·0019	—	—	—	—
1½	—	325	39·5	8·7	2·63	1	0·21	0·065	0·024	0·011	0·0051	—	—	—	—
2	—	1530	186	41	12·4	4·7	1	0·36	0·115	0·05	0·024	0·013	0·0074	—	—
2½	—	—	610	134	40·6	15·4	3·27	1	0·375	0·165	0·079	0·042	0·024	0·0092	—
3	—	—	—	357	108	41	8·7	2·67	1	0·44	0·21	0·112	0·065	0·024	0·0053
3½	—	—	—	810	243	93	19·5	6·05	2·27	1	0·48	0·255	0·147	0·056	0·012
4	—	—	—	—	514	195	41	12·6	4·74	2·09	1	0·533	0·308	0·116	0·025
4½	—	—	—	—	—	365	78	23·7	8·9	3·93	1·88	1	0·58	0·217	0·047
5	—	—	—	—	—	630	134	41	15·4	6·8	3·24	1·72	1	0·376	0·082
6	—	—	—	—	—	1680	356	109	41	18	8·62	4·6	2·66	1	0·22
8	—	—	—	—	—	—	1640	500	188	83	39·6	21	12	4·6	1
	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16

TABLE XXV.

HYDRAULIC MEMORANDA.

1 Imperial gallon of water	= 277·274 cubic inches
1 cubic foot of water	= 62·37 lbs
1 ,, inch ,, ...	= 0·036 ,,
A column of water 1 inch square and 1 foot high	= 0·434 ,,
A column of water 1 inch diameter and 1 foot high	= 0·34 ,,

TABLE XXV.—*continued.*

The capacity of a 1-foot cube	=	6·232 Imperial gallons.
The capacity of a tube 1 inch square and 1 foot long	=	0·0434 ,, ,,
The capacity of a tube 1 inch diameter and 1 foot long	=	0·034 ,, ,,
The capacity of a tube 1 foot diameter and 1 foot long	=	4·9 ,, ,,
The capacity of a sphere 1 foot diameter	=	3·263 ,, ,,
1 cubic foot sea water	=	64·001 lb.
1 ,, inch ,,	=	0·037 ,,
1 Imperial gallon	=	1·2 American gallon.
1 American ,,	=	0·83 Imperial ,,
,, ,,	=	231 cubic inches.
1 cubic foot of water	=	7·48 American gallons.
1 Imperial gallon	=	4·543 litres.
1 American ,,	=	3·8 ,,
1 cubic foot ,,	=	28·375 ,,
1 litre of water	=	0·22 Imperial gallon.
1 ,, ,,	=	0·264 American ,,
1 ,, ,,	=	61 cubic inches.
1 ,, ,,	=	0·0353 cubic foot.
1 cubic metre of water	=	220 Imperial gallons.
1 ,, ,,	=	264 American ,,

TABLE XXVI.

WEIGHT OF A CUBIC FOOT OF WATER AT DIFFERENT TEMPERATURES.

Temp. deg. F.	Weight lb. per cubic ft.	Temp. deg. F.	Weight lb. per cubic ft.	Temp. deg. F.	Weight lb. per cubic ft.
32	62·42	110	61·87	190	60·31
35	62·42	115	61·81	195	60·2
40	62·42	120	61·71	200	60·08
45	62·42	125	61·65	205	59·93
50	62·41	130	61·56	210	59·82
55	62·39	135	61·47	215	59·64
60	62·37	140	61·38	220	59·58
65	62·34	145	61·29	230	59·31
70	62·31	150	61·2	240	59·03
75	62·27	155	61·1	250	58·75
80	62·23	160	60·99	260	58·46
85	62·18	165	60·84	270	58·17
90	62·13	170	60·78	280	57·88
95	62·07	175	60·66	290	57·58
100	62·02	180	60·55	300	57·26
105	61·96	185	60·43	400	53·63

TABLE XXVII.

WEIGHT OF A SQUARE FOOT OF DIFFERENT METALS, FROM $\frac{1}{16}$ INCH TO 1 INCH THICK IN POUNDS.

Thickness, Inch.	Wrought iron.	Cast iron.	Steel.	Copper.	Zinc.	Tin.	Lead.
$\frac{1}{16}$	2·5	2·3	2·6	2·9	2·3	2·4	3·7
$\frac{1}{8}$	5·0	4·7	5·1	5·8	4·7	4·8	7·4
$\frac{3}{16}$	7·5	7·0	7·6	8·7	7·0	7·2	11·2
$\frac{1}{4}$	10·0	9·4	10·2	11·6	9·4	9·6	14·9
$\frac{5}{16}$	12·5	11·7	12·8	14·5	11·7	12·0	18·6
$\frac{3}{8}$	15·0	14·1	15·3	17·2	14·0	14·4	22·3
$\frac{7}{16}$	17·5	16·4	17·9	20·0	16·4	16·8	26·0
$\frac{1}{2}$	20·0	18·7	20·4	22·9	18·6	19·3	29·7
$\frac{9}{16}$	22·5	21·1	23·0	25·7	21·0	21·7	33·4
$\frac{5}{8}$	25·0	23·5	25·5	28·6	23·4	24·1	37·1
$\frac{11}{16}$	27·5	25·8	28·1	31·4	25·7	26·5	40·9
$\frac{3}{4}$	30·0	28·1	30·6	34·3	28·0	28·9	44·6
$\frac{13}{16}$	32·5	30·5	33·2	37·2	30·4	31·3	48·3
$\frac{7}{8}$	35·0	32·8	35·7	40·0	32·7	33·7	52·0
$\frac{15}{16}$	37·5	35·2	38·3	42·9	35·1	36·1	55·7
1	40·0	37·5	40·8	45·8	37·4	38·5	59·4

TABLE XXVIII.

WEIGHT OF ONE SQUARE FOOT OF METALS.

New standard wire gauges. No.	Wrought iron. Lb.	Steel. Lb.	Copper. Lb.	Tin. Lb.	Zinc. Lb.	Lead. Lb.
1	11·92	12·24	13·7	11·32	11·23	17·75
2	10·97	11·26	12·63	10·42	10·35	16·45
3	10·02	10·29	11·53	9·52	9·45	15·03
4	9·22	9·47	10·61	8·76	8·70	13·83
5	8·43	8·66	9·70	8·01	7·95	12·64
6	7·63	7·84	9·78	7·25	7·20	11·44
7	6·86	7·04	7·90	6·62	6·48	10·20
8	6·36	6·53	7·32	6·04	6·00	9·54
9	5·72	6·13	6·58	5·43	5·40	8·58
10	5·08	5·22	5·85	4·83	4·80	7·62
11	4·61	4·73	5·31	4·38	4·35	6·91
12	4·13	4·24	4·75	3·92	3·89	6·20
13	3·66	3·76	4·21	3·48	3·45	5·49
14	3·18	3·26	3·66	3·02	3·00	4·77
15	2·86	2·94	3·30	2·72	2·70	4·30
16	2·54	2·60	2·92	2·41	2·40	3·81
17	2·14	2·19	2·46	2·03	2·02	3·21

TABLE XXVIII.—*continued.*

New standard wire gauges. No.	Wrought iron. Lb.	Steel. Lb.	Copper. Lb.	Tin. Lb.	Zinc. Lb.	Lead. Lb.
18	1·91	1·96	2·20	1·81	1·80	2 86
19	1·59	1·63	1·83	1·51	1·49	2·38
20	1·43	1·47	1·64	1·36	1·35	2·14
21	1·28	1·31	1·47	1·22	1·24	1·92
22	1·11	1·14	1·28	1·05	1·04	1·66
23	0·95	0·97	1·09	0·90	0·89	1·43
24	0·87	0·89	1·00	0·83	0·82	1·30
25	0·79	0·81	0·91	0 75	0·74	1·18
26	0·71	0·73	0·82	0·67	0·67	1·06
27	0·65	0·67	0·75	0·62	0·62	0·97
28	0·58	0·60	0·66	0·55	0·54	0·87
29	0·54	0·55	0·62	0·51	0·50	0·81
30	0·50	0·51	0·58	0·47	0·47	0·75

TABLE XXIX.

WEIGHT OF CAST-IRON PIPES PER LINEAL FOOT.

Bore. Inches.	Thickness of metal.							
	$\frac{1}{4}$ in.	$\frac{3}{8}$	$\frac{1}{2}$ in.	$\frac{5}{8}$ in.	$\frac{3}{4}$ in.	$\frac{7}{8}$ in.	1 in.	$1\frac{1}{8}$ in.
$1\frac{1}{2}$	4·3	6·9	9·8	13·0	—	—	—	—
2	5·5	8·7	12·3	16·1	—	—	—	—
3	8·0	12·4	17·1	22·2	—	—	—	—
4	10·4	16·1	22·1	28·3	34·9	—	—	—
5	12·9	19·8	26·9	34·4	42·3	—	—	—
6	15·3	23·4	31·9	40·6	49·7	—	—	—
7	—	27·1	36·8	46·7	56·8	—	—	—
8	—	30·8	41·6	52·8	64·3	—	—	—
9	—	34·4	46·0	58·9	71·7	—	—	—
10	—	—	51·4	65·1	79·0	93·3	—	—
11	—	—	56·4	71·0	86·4	101·8	—	—
12	—	—	—	77·3	93·7	110·4	127·4	—
14	—	—	—	89·6	108·4	127·5	147·0	—
15	—	—	—	—	115·7	136·1	156·8	177·7
16	—	—	—	—	123·1	144·7	166·6	188·7
18	—	—	—	—	137·9	161·8	186·2	210·8

The above weights are for plain pipe ends.

TABLE XXX.

WIRE AND PLATE GAUGES.

No.	Equivalent diameter or thickness in the fraction of an inch.			No.	Equivalent diameter or thickness in the fraction of an inch.		
	New standard wire gauge	Birmingham wire gauge.	American wire gauge.		New standard wire gauge.	Birmingham wire gauge.	American wire gauge.
7/0	0·500	—	—	21	0·032	0·032	0·0284
6/0	0·464	—	—	22	0·028	0·03	0·0253
5/0	0·432	—	—	23	0·024	0·025	0·022
0000	0·400	0·454	0·46	24	0·022	0·022	0·02
000	0·372	0·425	0·409	25	0·02	0·02	0·018
00	0·348	0·38	0·365	26	0·018	0·018	0·016
0	0·324	0·34	0·325	27	0·016	0·016	0·014
1	0·3	0·3	0·289	28	0·014	0·014	0·0122
2	0·276	0·284	0·257	29	0·013	0·013	0·011
3	0·252	0·259	0·229	30	0·012	0·012	0·01
4	0·232	0·238	0·204	31	0·011	0·01	0·009
5	0·212	0·22	0·182	32	0·0108	0·009	0·008
6	0·192	0·203	0·162	33	0·01	0·008	0·007
7	0·176	0·18	0·144	34	0·009	0·007	0·006
8	0·16	0·165	0·128	35	0·008	0·005	0·0056
9	0·144	0·148	0·114	36	0·0076	0·004	0·005
10	0·128	0·134	0·102	37	0·0068	—	0·0044
11	0·116	0·12	0·09	38	0·006	—	0·004
12	0·104	0·109	0·08	39	0·005	—	0·0036
13	0·092	0·095	0·072	40	0·0048	—	0·0032
14	0·08	0·083	0·064	41	0·0044	—	—
15	0·072	0·072	0·057	42	0·004	—	—
16	0·064	0·065	0·05	43	0·0036	—	—
17	0·056	0·058	0·045	44	0·0032	—	—
18	0·048	0·049	0·04	45	0·0028	—	—
19	0·04	0·042	0·036	46	0·0024	—	—
20	0·036	0·035	0·032				

TABLE XXXI.

COMMON LOGARITHMS.

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170						4	9	13	17	21	25	30	34	38
						0212	0253	0294	0334	0374	4	8	12	16	20	24	28	32	37
11	0414	0453	0492	0531	0569						4	8	12	15	19	23	27	31	35
						0607	0645	0682	0719	0755	4	7	11	15	19	22	26	30	33
12	0792	0828	0864	0899	0934	0969					3	7	11	14	18	21	25	28	32
							1004	1038	1072	1106	3	7	10	14	17	20	24	27	31
13	1139	1173	1206	1239	1271						3	7	10	13	16	20	23	26	30
						1303	1335	1367	1399	1430	3	7	10	12	16	19	22	25	29
14	1461	1492	1523	1553							3	6	9	12	15	18	21	24	28
					1584	1614	1644	1673	1703	1732	3	6	9	12	15	17	20	23	26
15	1761	1790	1818	1847	1875	1903					3	6	9	11	14	17	20	23	26
							1931	1959	1987	2014	3	5	8	11	14	16	19	22	25
16	2041	2068	2095	2122	2148						3	5	8	11	14	16	19	22	24
						2175	2201	2227	2253	2279	3	5	8	10	13	15	18	21	23
17	2304	2330	2355	2380	2405	2430					3	5	8	10	13	15	18	20	23
							2455	2480	2504	2529	2	5	7	10	12	15	17	19	22
18	2553	2577	2601	2625	2648						2	5	7	9	12	14	16	19	21
						2672	2695	2718	2742	2765	2	5	7	9	11	14	16	18	21
19	2788	2810	2833	2856	2878						2	4	7	9	11	13	16	18	20
						2900	2923	2945	2967	2989	2	4	6	8	11	13	15	17	19
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8

TABLE XXXI.—*continued.*

COMMON LOGARITHMS.

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
51	7076	[illegible]	[illegible]	7101	7110	[illegible]	[illegible]	[illegible]	[illegible]	7152	[illegible]	[illegible]	[illegible]	[illegible]	4	[illegible]	[illegible]	7	[illegible]
52	7160	[illegible]	[illegible]	[illegible]	[illegible]	7202	7210	7218	7226	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	4	[illegible]	[illegible]	7	7
53	[illegible]	[illegible]	[illegible]	7267	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	4	[illegible]	[illegible]	[illegible]	[illegible]
54	[illegible]	[illegible]	[illegible]	7348	[illegible]	7364	[illegible]	[illegible]	[illegible]	[illegible]	1	[illegible]	[illegible]	[illegible]	4	[illegible]	[illegible]	[illegible]	7
55	7404	7412	[illegible]	[illegible]	[illegible]	7443	[illegible]	[illegible]	[illegible]	7474	1	2	[illegible]	3	4	[illegible]	5	6	[illegible]
56	[illegible]	7490	[illegible]	[illegible]	[illegible]	[illegible]	7528	7536	7543	7551	1	[illegible]	[illegible]	3	4	5	5	[illegible]	7
57	[illegible]	7566	[illegible]	[illegible]	[illegible]	[illegible]	7604	7612	7619	7627	[illegible]	[illegible]	[illegible]	3	4	5	[illegible]	6	7
58	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	7672	7679	7686	7694	7701	1	1	2	[illegible]	4	[illegible]	[illegible]	[illegible]	7
59	[illegible]	[illegible]	7723	7731	7738	[illegible]	7752	7760	7767	7774	1	1	[illegible]	[illegible]	[illegible]	4	[illegible]	[illegible]	[illegible]
60	[illegible]	[illegible]	[illegible]	7803	[illegible]	7818	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	2	3	4	4	[illegible]	6	6
61	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	7910	[illegible]	1	1	2	[illegible]	4	4	[illegible]	6	6
62	[illegible]	[illegible]	[illegible]	7945	7952	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	[illegible]	3	[illegible]	4	[illegible]	[illegible]	[illegible]
63	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	2	3	[illegible]	4	5	6	6
64	8062	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	2	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
65	8129	[illegible]	[illegible]	[illegible]	8156	8162	[illegible]	[illegible]	[illegible]	[illegible]	1	1	[illegible]	[illegible]	[illegible]	4	5	5	6
66	[illegible]	[illegible]	8209	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	2	[illegible]	[illegible]	4	5	5	6
67	[illegible]	[illegible]	[illegible]	8280	8287	8293	8299	8306	8312	8319	1	1	2	[illegible]	[illegible]	4	5	5	6
68	[illegible]	[illegible]	8338	[illegible]	[illegible]	[illegible]	8363	8370	8376	8382	[illegible]	1	[illegible]	3	3	4	4	[illegible]	[illegible]
69	[illegible]	[illegible]	8401	8407	[illegible]	8420	8426	8432	8439	8445	1	1	2	[illegible]	3	4	4	[illegible]	[illegible]
70	[illegible]	[illegible]	[illegible]	8470	8476	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	[illegible]	[illegible]	[illegible]	3	4	4	5	6
71	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	[illegible]	2	[illegible]	4	4	[illegible]	[illegible]
72	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	8603	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	[illegible]	[illegible]	3	4	4	[illegible]	[illegible]
73	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	[illegible]	[illegible]	3	4	4	[illegible]	[illegible]
74	[illegible]	[illegible]	8704	8710	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	2	2	3	4	[illegible]	5	5
75	[illegible]	[illegible]	8762	8768	8774	[illegible]	8785	8791	8797	8802	1	1	2	2	[illegible]	[illegible]	4	5	5
76	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	8837	[illegible]	[illegible]	[illegible]	[illegible]	1	1	2	[illegible]	[illegible]	[illegible]	4	[illegible]	5
77	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	2	[illegible]	[illegible]	[illegible]	4	4	5
78	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	2	[illegible]	[illegible]	3	4	4	5
79	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	2	[illegible]	3	3	4	4	5
80	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	9058	9063	9069	9074	9079	[illegible]	[illegible]	[illegible]	2	3	3	4	4	5
81	[illegible]	[illegible]	[illegible]	9101	[illegible]	9112	[illegible]	[illegible]	[illegible]	[illegible]	1	[illegible]	[illegible]	2	[illegible]	3	4	4	[illegible]
82	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	9165	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	[illegible]	[illegible]	[illegible]	3	4	[illegible]	[illegible]
83	[illegible]	9196	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3	[illegible]	4	4	[illegible]
84	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	9269	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	2	2	3	[illegible]	[illegible]	4	[illegible]
85	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	2	2	[illegible]	3	4	4	[illegible]
86	[illegible]	[illegible]	[illegible]	9360	9365	9370	[illegible]	[illegible]	[illegible]	[illegible]	1	1	2	[illegible]	3	3	4	4	[illegible]
87	[illegible]	[illegible]	[illegible]	9410	9415	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0	1	1	2	2	3	3	4	4
88	9445	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	[illegible]	2	[illegible]	[illegible]	4	4
89	[illegible]	[illegible]	[illegible]	9509	9513	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	1	[illegible]	2	3	3	4	4
90	9542	9547	[illegible]	[illegible]	[illegible]	9566	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1	2	2	3	3	[illegible]	4
91	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0	1	1	[illegible]	2	3	3	[illegible]	4
92	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	9661	[illegible]	[illegible]	[illegible]	[illegible]	0	1	1	[illegible]	2	[illegible]	[illegible]	4	4
93	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3	3	[illegible]	[illegible]
94	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	2	[illegible]	[illegible]	4	4
95	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	9800	[illegible]	[illegible]	[illegible]	[illegible]	0	1	1	[illegible]	[illegible]	[illegible]	[illegible]	4	4
96	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3	3	4	4
97	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0	1	1	[illegible]	[illegible]	3	3	4	4
98	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0	1	1	[illegible]	[illegible]	[illegible]	[illegible]	4	4
99	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

TABLE XXXII.

ANTILOGARITHMS.

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·00	1000	1002	1005	1007	1009	1012	1014	1016	1019	1021	0	0	1	1	1	1	2	2	2
·01	1023	1026	1028	1030	1033	1035	1038	1040	1042	1045	0	0	1	1	1	1	2	2	2
·02	1047	1050	1052	1054	1057	1059	1062	1064	1067	1069	0	0	1	1	1	1	2	2	2
·03	1072	1074	1076	1079	1081	1084	1086	1089	1091	1094	0	0	1	1	1	1	2	2	2
·04	1096	1099	1102	1104	1107	1109	1112	1114	1117	1119	0	1	1	1	1	2	2	2	2
·05	1122	1125	1127	1130	1132	1135	1138	1140	1143	1146	0	1	1	1	1	2	2	2	2
·06	1148	1151	1153	1156	1159	1161	1164	1167	1169	1172	0	1	1	1	1	2	2	2	2
·07	1175	1178	1180	1183	1186	1189	1191	1194	1197	1199	0	1	1	1	1	2	2	2	2
·08	1202	1205	1208	1211	1213	1216	1219	1222	1225	1227	0	1	1	1	1	2	2	2	3
·09	1230	1233	1236	1239	1242	1245	1247	1250	1253	1256	0	1	1	1	1	2	2	2	3
·10	1259	1262	1265	1268	1271	1274	1276	1279	1282	1285	0	1	1	1	1	2	2	2	3
·11	1288	1291	1294	1297	1300	1303	1306	1309	1312	1315	0	1	1	1	2	2	2	2	3
·12	1318	1321	1324	1327	1330	1334	1337	1340	1343	1346	0	1	1	1	2	2	2	2	3
·13	1349	1352	1355	1358	1361	1365	1368	1371	1374	1377	0	1	1	1	2	2	2	3	3
·14	1380	1384	1387	1390	1393	1396	1400	1403	1406	1409	0	1	1	1	2	2	2	3	3
·15	1413	1416	1419	1422	1426	1429	1432	1435	1439	1442	0	1	1	1	2	2	2	3	3
·16	1445	1449	1452	1455	1459	1462	1466	1469	1472	1476	0	1	1	1	2	2	2	3	3
·17	1479	1483	1486	1489	1493	1496	1500	1503	1507	1510	0	1	1	1	2	2	2	3	3
·18	1514	1517	1521	1524	1528	1531	1535	1538	1542	1545	0	1	1	1	2	2	2	3	3
·19	1549	1552	1556	1560	1563	1567	1570	1574	1578	1581	0	1	1	1	2	2	3	3	3
·20	1585	1589	1592	1596	1600	1603	1607	1611	1614	1618	0	1	1	1	2	2	3	3	3
·21	1622	1626	1629	1633	1637	1641	1644	1648	1652	1656	0	1	1	2	2	2	3	3	3
·22	1660	1663	1667	1671	1675	1679	1683	1687	1690	1694	0	1	1	2	2	2	3	3	3
·23	1698	1702	1706	1710	1714	1718	1722	1726	1730	1734	0	1	1	2	2	2	3	3	4
·24	1738	1742	1746	1750	1754	1758	1762	1766	1770	1774	0	1	1	2	2	2	3	3	4
·25	1778	1782	1786	1791	1795	1799	1803	1807	1811	1816	0	1	1	2	2	2	3	3	4
·26	1820	1824	1828	1832	1837	1841	1845	1849	1854	1858	0	1	1	2	2	3	3	3	4
·27	1862	1866	1871	1875	1879	1884	1888	1892	1897	1901	0	1	1	2	2	3	3	3	4
·28	1905	1910	1914	1919	1923	1928	1932	1936	1941	1945	0	1	1	2	2	3	3	4	4
·29	1950	1954	1959	1963	1968	1972	1977	1982	1986	1991	0	1	1	2	2	3	3	4	4
·30	1995	2000	2004	2009	2014	2018	2023	2028	2032	2037	0	1	1	2	2	3	3	4	4
·31	2042	2046	2051	2056	2061	2065	2070	2075	2080	2084	0	1	1	2	2	3	3	4	4
·32	2089	2094	2099	2104	2109	2113	2118	2123	2128	2133	0	1	1	2	2	3	3	4	4
·33	2138	2143	2148	2163	2158	2163	2168	2173	2178	2183	0	1	1	2	2	3	3	4	4
·34	2188	2193	2198	2203	2208	2213	2218	2223	2228	2234	1	1	2	2	3	3	4	4	5
·35	2239	2244	2249	2254	2259	2265	2270	2275	2280	2286	1	1	2	2	3	3	4	4	5
·36	2291	2296	2301	2307	2312	2317	2323	2328	2333	2339	1	1	2	2	3	3	4	4	5
·37	2344	2350	2355	2360	2366	2371	2377	2382	2388	2393	1	1	2	2	3	3	4	4	5
·38	2399	2404	2410	2415	2421	2427	2432	2438	2443	2449	1	1	2	2	3	3	4	4	5
·39	2455	2460	2466	2472	2477	2483	2489	2495	2500	2506	1	1	2	2	3	3	4	5	5
·40	2512	2518	2523	2529	2535	2541	2547	2553	2559	2564	1	1	2	2	3	4	4	5	5
·41	2570	2576	2582	2588	2594	2600	2606	2612	2618	2624	1	1	2	2	3	4	4	5	5
·42	2630	2636	2642	2649	2655	2661	2667	2673	2679	2685	1	1	2	2	3	4	4	5	6
·43	2692	2698	2704	2710	2716	2723	2729	2735	2742	2748	1	1	2	3	3	4	4	5	6
·44	2754	2761	2767	2773	2780	2786	2793	2799	2805	2812	1	1	2	3	3	4	4	5	6
·45	2818	2825	2831	2838	2844	2851	2858	2864	2871	2877	1	1	2	3	3	4	5	5	6
·46	2884	2891	2897	2904	2911	2917	2924	2931	2938	2944	1	1	2	3	3	4	5	5	6
·47	2951	2958	2965	2972	2979	2985	2992	2999	3006	3013	1	1	2	3	3	4	5	5	6
·48	3020	3027	3034	3041	3048	3055	3062	3069	3076	3083	1	1	2	3	4	4	5	6	6
·49	3090	3097	3105	3112	3119	3126	3133	3141	3148	3155	1	1	2	3	4	4	5	6	6

ANTILOGARITHMS.

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·50	3162	3170	3177	3184	3192	3199	3206	3214	3221	3228	1	1	2	3	4	4	5	6	7
·51	3236	3243	3251	3258	3266	3273	3281	3289	3296	3304	1	2	2	3	4	5	5	6	7
·52	3311	3319	3327	3334	3342	3350	3357	3365	3373	3381	1	2	2	3	4	5	5	6	7
·53	3388	3396	3404	3412	3420	3428	3436	3443	3451	3459	1	2	2	3	4	5	6	6	7
·54	3467	3475	3483	3491	3499	3508	3516	3524	3532	3540	1	2	2	3	4	5	6	6	7
·55	3548	3556	3565	3573	3581	3589	3597	3606	3614	3622	1	2	2	3	4	5	6	7	7
·56	3631	3639	3648	3656	3664	3673	3681	3690	3698	3707	1	2	3	3	4	5	6	7	8
·57	3715	3724	3733	3741	3750	3758	3767	3776	3784	3793	1	2	3	3	4	5	6	7	8
·58	3802	3811	3819	3828	3837	3846	3855	3864	3873	3882	1	2	3	4	4	5	6	7	8
·59	3890	3899	3908	3917	3926	3936	3945	3954	3963	3972	1	2	3	4	5	5	6	7	8
·60	3981	3990	3999	4009	4018	4027	4036	4046	4055	4064	1	2	3	4	5	6	6	7	8
·61	4074	4083	4093	4102	4111	4121	4130	4140	4150	4159	1	2	3	4	5	6	7	8	9
·62	4169	4178	4188	4198	4207	4217	4227	4236	4246	4256	1	2	3	4	5	6	7	8	9
·63	4266	4276	4285	4295	4305	4315	4325	4335	4345	4355	1	2	3	4	5	6	7	8	9
·64	4365	4375	4385	4395	4406	4416	4426	4436	4446	4457	1	2	3	4	5	6	7	8	9
·65	4467	4477	4487	4498	4508	4519	4529	4539	4550	4560	1	2	3	4	5	6	7	8	9
·66	4571	4581	4592	4603	4613	4624	4634	4645	4656	4667	1	2	3	4	5	6	7	9	10
·67	4677	4688	4699	4710	4721	4732	4742	4753	4764	4775	1	2	3	4	5	7	8	9	10
·68	4786	4797	4808	4819	4831	4842	4853	4864	4875	4887	1	2	3	4	6	7	8	9	10
·69	4898	4909	4920	4932	4943	4955	4966	4977	4989	5000	1	2	3	5	6	7	8	9	10
·70	5012	5023	5035	5047	5058	5070	5082	5093	5105	5117	1	2	4	5	6	7	8	9	11
·71	5129	5140	5152	5164	5176	5188	5200	5212	5224	5236	1	2	4	5	6	7	8	10	11
·72	5248	5260	5272	5284	5297	5309	5321	5333	5346	5358	1	2	4	5	6	7	9	10	11
·73	5370	5383	5395	5408	5420	5433	5445	5458	5470	5483	1	3	4	5	6	8	9	10	11
·74	5495	5508	5521	5534	5546	5559	5572	5585	5598	5610	1	3	4	5	6	8	9	10	12
·75	5623	5636	5649	5662	5675	5689	5702	5715	5728	5741	1	3	4	5	7	8	9	10	12
·76	5754	5768	5781	5794	5808	5821	5834	5848	5861	5875	1	3	4	5	7	8	9	11	12
·77	5888	5902	5916	5929	5943	5957	5970	5984	5998	6012	1	3	4	5	7	8	10	11	12
·78	6026	6039	6053	6067	6081	6095	6109	6124	6138	6152	1	3	4	6	7	8	10	11	13
·79	6166	6180	6194	6209	6223	6237	6252	6266	6281	6295	1	3	4	6	7	9	10	11	13
·80	6310	6324	6339	6353	6368	6383	6397	6412	6427	6442	1	3	4	6	7	9	10	12	13
·81	6457	6471	6486	6501	6516	6531	6546	6561	6577	6592	2	3	5	6	8	9	11	12	14
·82	6607	6622	6637	6653	6668	6683	6699	6714	6730	6745	2	3	5	6	8	9	11	12	14
·83	6761	6776	6792	6808	6823	6839	6855	6871	6887	6902	2	3	5	6	8	9	11	13	14
·84	6918	6934	6950	6966	6982	6998	7015	7031	7047	7063	2	3	5	6	8	10	11	13	15
·85	7079	7096	7112	7129	7145	7161	7178	7194	7211	7228	2	3	5	7	8	10	12	13	15
·86	7244	7261	7278	7295	7311	7328	7345	7362	7379	7396	2	3	5	7	8	10	12	13	15
·87	7413	7430	7447	7464	7482	7499	7516	7534	7551	7568	2	3	5	7	9	10	12	14	16
·88	7586	7603	7621	7638	7656	7674	7691	7709	7727	7745	2	4	5	7	9	11	12	14	16
·89	7762	7780	7798	7816	7834	7852	7870	7889	7907	7925	2	4	5	7	9	11	13	14	16
·90	7943	7962	7980	7998	8017	8035	8054	8072	8091	8110	2	4	6	7	9	11	13	15	17
·91	8128	8147	8166	8185	8204	8222	8241	8260	8279	8299	2	4	6	8	9	11	13	15	17
·92	8318	8337	8356	8375	8395	8414	8433	8453	8472	8492	2	4	6	8	10	12	14	15	17
·93	8511	8531	8551	8570	8590	8610	8630	8650	8670	8690	2	4	6	8	10	12	14	16	18
·94	8710	8730	8750	8770	8790	8810	8831	8851	8872	8892	2	4	6	8	10	12	14	16	18
·95	8913	8933	8964	8974	8995	9016	9036	9057	9078	9099	2	4	6	8	10	12	15	17	19
·96	9120	9141	9162	9183	9204	9226	9247	9268	9290	9311	2	4	6	8	11	13	15	17	19
·97	9333	9354	9376	9397	9419	9441	9462	9484	9506	9528	2	4	7	9	11	13	15	17	20
·98	9550	9572	9594	9616	9638	9661	9683	9705	9727	9750	2	4	7	9	11	13	16	18	20
·99	9772	9795	9817	9840	9863	9886	9908	9931	9954	9977	2	5	7	9	11	14	16	18	20

INDEX

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